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SOUND AND VIBRATION IN MARINE GEARING, A CONTINUATION.

By AUSTIN KUHNS, Member.*

This article is the second by Mr. Kuhns dealing with the problem of noise production in marine gearing, and its control. It should be read as a companion piece of the first which appeared in the August, 1937, issue of the JOURNAL.

In the introduction to the first article, it was explained that the acuteness of the noise problem brought about by the use of higher steam pressures, higher machinery speeds, and more compact machinery layouts in marine installations, is causing the most earnest study of it by all machinery designers and operators. Mr. Kuhns' long association with marine gear manufacture permits him to speak with authority on the subject.

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The first article contained a statement of the problem, and a general discussion of the methods and apparatus at hand to combat it. This article follows up with a study of the data which was taken in specific cases, and draws certain conclusions which may be incorporated in future design. The Farrel-Birmingham Company is to be congratulated on the progressive steps it has taken in the attempt to clarify and isolate the confusing factors which so often combine to turn a normal gear assembly into an irritating noise-maker.

THE APPROACH TO ACTUAL EXPERIENCE.

We now believe that our understanding of this problem of sound and vibration in marine gearing is clear and, as our foregoing discussion explains, is capable of very definite approach. With this as our basis, therefore, we have undertaken actual investigation, experimentation and the preparation of specifications for routine manufacturing operations.

We have obtained sufficient information from specific measurements to enable us to formulate detailed directions for determining with accuracy the overall sound and vibration level of the finished reduction gear unit. We also know how to break up this overall level into a spectrum which permits a detailed analysis of the operating characteristics. Our results can be general or localized, depending upon how and where we place our microphone or vibration pick-up.

We have discovered new methods for locating during manufacture defective or harmful conditions in individual parts and groups of parts which go to make up the completed reduction unit. We have become decidedly optimistic over the possibilities for diagnosing and correcting these injurious conditions while it is still possible to make changes in design or manufacturing methods. We firmly expect that the diligent use of such inspection technique will bring about noteworthy improvement in the operating performance of propulsion gearing.

STUDIES OF STATIONARY INSTALLATIONS.

The first specific step in the approach to the problem has been the work of certain investigators with sound and vibration measuring equipment applied directly to operating gear sets, especially those

in stationary service. For purposes of comparison, standard schedules have been adopted and the necessary assumptions carefully considered. The results usually have been limited to overall sound levels and certain attempts at spectrum analysis.

AT U. S. RUBBER CO., NAUGATUCK, CONN.

A typical examination of this type was carried out by the writer's collaborators* at the plant of the U. S. Rubber Company at Naugatuck, Conn., where preliminary inspection showed an unusually wide variation in operating characteristics between the group of gear sets of our manufacture already installed there. Obviously, all measurements were taken during working hours and the usual difficulties were encountered with interference and distortion from the surrounding machinery and the adjacent walls and stock piles. Resonance and reverberation and actual foundation vibration, of course, were present and have required arbitrary correction of the actual readings. The results, however, as tabulated below are indicative of what can be expected from such investigations of actual operating conditions.

AVERAGE NOISE INTENSITIES.

	decibels
D-8 gear unit	82
D-10 gear unit	90
Large gear unit in basement	95
Gear unit in gallery	80
Gear unit of West roll assembly	75
Gear unit of East roll assembly	82

AVERAGE SURFACE VIBRATION.

	decibels
D-8 gear unit	92
D-10 gear unit	102
Large gear unit in basement	94
Gear unit in gallery	83
Gear unit of West roll assembly	75
Gear unit of East roll assembly	82

* Dr. E. E. Free and Associates of New York City.

AVERAGE NATURAL ROOM AND MOTOR NOISE.

	decibels
Location of—D-8 gear unit	81
Location of—D-10 gear unit	87
Location of—Large gear unit in basement	95
Location of—Gear unit in gallery	83
Location of—Gear unit of West roll assembly	76
Location of—Gear unit of East roll assembly	83

Examination of these averages discloses the wide variation which can exist between sound and vibration levels of reduction gear units of the same manufacture and in the same class of service. Whereas some of this is the result of the respective locations and inherent characteristics in design, much is due to differences in actual quality of manufacture.

FREQUENCY ANALYSES.

The average sound levels were supplemented with careful analyses of the spectrum through the use of filters previously described. The results are of sufficient importance for tabulation herein.

NOISE MEASUREMENTS.

Frequency bands. Cycles per Second	D-8 Unit	D-10 Unit	Unit in Basement	Unit in Gallery	Unit on West Rolls	Unit on East Rolls
40–64	52	50	72	52	46	57
64–128	62	68	86	69	60	72
128–256	71	82	88	78	68	78
256–512	76	87	88	77	70	77
512–1024	76	86	90	75	66	73
1024–2048	72	81	87	71	66	71
2048–4096	62	68	79	67	64	65
4096–8192	55	59	69	64	58	60

These figures are interesting in the uniformity of the curve of sound intensity versus frequency and the occurrence of the peak at 256-512 cycles per second.

VIBRATION STUDIES.

The vibration analyses with figures expressed in terms of equivalent sound intensities in air are given below.

SURFACE VIBRATION.

Frequency Bands. Cycles per Second	D-8 Unit	D-10 Unit	Unit in Basement	Unit in Gallery	Unit on West Rolls	Unit on East Rolls
40–64	61	61	74	70	49	76
64–128	67	86	89	73	64	79
128–256	83	98	88	73	68	80
256–512	86	94	86	78	66	79
512–1024	85	88	84	77	62	79
1024–2048	75	80	77	71	58	80
2048–4096	70	73	75	69	60	80
4096–8192	64	59	74	67	44	75

The results obtained from these measurements follow the general pattern of the noise spectrum. Such variations as exist are logically explained by the natural differences which are found in air-borne and machine-borne vibrations.

NATURAL ROOM NOISES.

Natural room and motor noises were examined and the readings are tabulated herewith.

NATURAL ROOM NOISE LEVEL.

Frequency Bands. Cycles per Second	D-8 Unit	D-10 Unit	Unit in Basement	Unit in Gallery	Unit on West Rolls	Unit on East Rolls
40–64	45	56	60	60	46	53
64–128	65	72	76	63	60	58
128–256	71	78	82	77	68	73
256–512	75	82	88	83	70	74
512–1024	76	80	91	81	66	75
1024–2048	72	74	90	79	66	76
2048–4096	62	69	83	72	64	76
4096–8192	55	62	74	65	58	69

Experience shows that there are an infinite number of positions for the microphone from which correspondingly varying results can be obtained when measuring natural room, motor and related noise levels. The confusion which this situation can create and the difficulty in determining the effects which these "outside" noises can have on the studies of the gear units themselves becomes a real factor for consideration.

FORMULATION OF GUARANTEE.

However, using this investigation as our basis and carefully considering all of the results obtained (only the summaries have been tabulated here) we drew up a set of standard specifications which, in effect, stated our guarantee for gear units of this general type. We quote this as a matter of interest and information.

"The sound energy flux emitted by the gear unit averaged in all directions from the unit at a distance of 1 meter from the nearest point on the surface of the unit, shall not exceed 87 decibels above 10^{-16} watt per square centimeter, the unit being operated at full load, on the equivalent of a non-vibrating and non-resonant base and in a non-reverberant room. The noise emitted shall not contain any single frequency tone greater in intensity than 5 decibels above the average noise level and in addition shall be free from irregular sounds, such as thumps and clicks."

No attempt was made to formulate an expression for vibration characteristics and intensities out of an appreciation of the complexity of the problem and its intimate relation to the structure on which the gear unit must be supported and to which it must be connected.

SPECIAL FULL-SCALE RESEARCH.

Recognition of the assumptions and limitations imposed with measurements under operating conditions and of the impossibility of determining with requisite accuracy the effects of the surroundings on the results obtained led to a series of comprehensive tests under conditions simulating the research laboratory.

Complete control over the operating conditions of a selected gear set was accomplished effectively by installing a companionate unit of opposite hand on the same test bed with connecting shafts between the respective rotating members. By twisting these connecting shafts and properly locking them to hold the live condition, all of the gear teeth and bearings in the selected gear set were loaded with the predetermined torques and in the correct directions. By rotating these members at the desired speed and by including the complete lubricating system and other auxiliaries, service conditions were duplicated very closely and with the advantage that full control was available at all times and changes in load and speed made possible easily and quickly.

SOUND-PROOF TEST ENCLOSURE.

To eliminate resonance, reverberation and vibration, the selected reduction gear unit was entirely enclosed in a specially constructed sound insulating and sound absorbing house. This consisted of walls constructed of two layers of porous fibre board with a four-inch dead air space between and three inches of rock wool and an inch of felt tufted on the inner side of the inside boarding. Real care was taken to fit the side walls around the protruding shafts and pipes and to cover with wool and felt the flooring, foundation structure and other possible sources of reverberation and resonance. The two doorways were effectively plugged with special door pieces of the same construction as the side walls and the complete enclosure was covered with a flat panel made in exactly the same way.

Preliminary examination of this structure showed a differential of fully 25 decibels between its interior and the exterior surroundings and very effective sound absorptive characteristics of the walls and ceiling.

INSTRUMENTS USED.

The instruments used in conducting this investigation have been described in the article which appeared in the August, 1937, issue of the JOURNAL. Included were, of course, the dynamic microphone and the audio-frequency vibration and crystal vibration pick-up, the sound meter and vibration meter, the frequency analyzer for separating the spectrum into eight bands and the cathode-ray oscilloscope.

LOADS AND SPEEDS APPLIED.

Twenty-five, fifty, seventy-five and one hundred per cent torque were the applied loads successively selected with a speed range from thirty to one hundred and fifteen per cent.

GENERAL RESULTS OBTAINED.

The results obtained from these tests were highly gratifying. The sounds emanating from the gear set were readily isolated and disturbance from external noises and interference from reverberation, vibration or resonance were reduced to negligible quantities.

The effectiveness of the sound-proof housing was proved by measurements taken while surrounding shop machinery was operating, although the actual work of examination was deferred until over week-ends when the external noises were at a minimum.

A brief examination of the summarized figures showed that the total sound level of this reduction gear unit when loaded with 100 per cent torque at 100 per cent rated speed averaged 85 decibels. The difference in level with change in torque between 25 per cent and 100 per cent was insignificant. The variation with increase in speed from 33 per cent to 115 per cent ran from 76 to 87 decibels.

FREQUENCY ANALYSES.

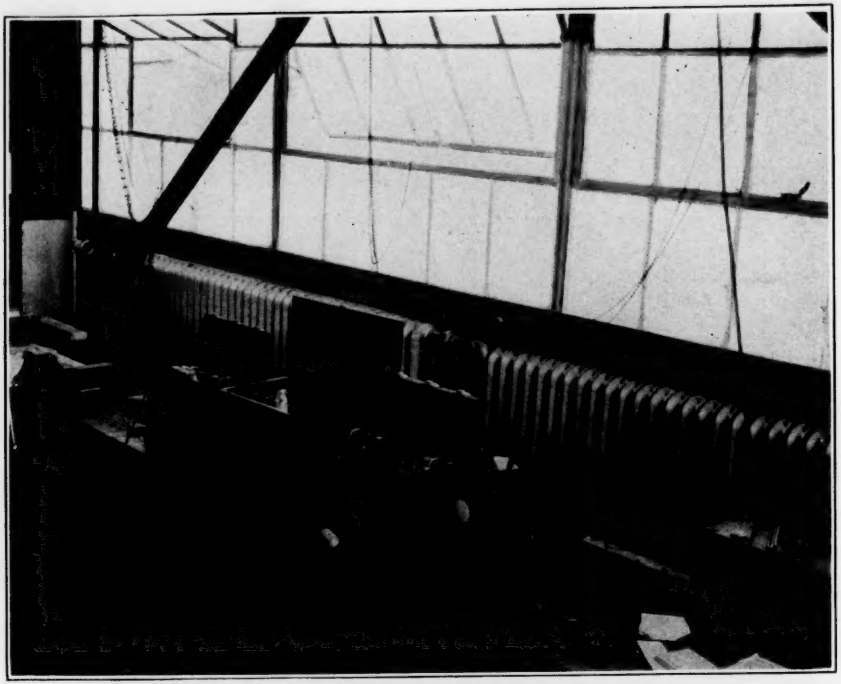
Frequency analyses at all speeds and torques tested showed a complex spectrum containing a large number of different frequencies with little difference in the bands between 64 cycles per second and 2048 cycles per second, below and above which the sound emitted was substantially less. Isolated peaks developed at the 128-256 cycle band and at 512-1024 cycles per second which it is believed reflect some mechanical characteristics of the unit itself.

GENERAL CONCLUSIONS.

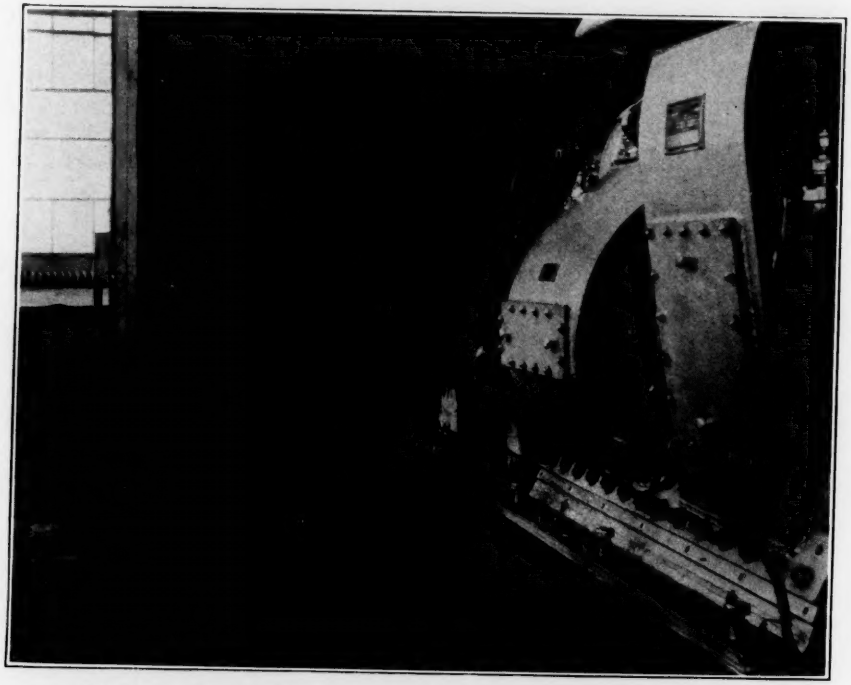
From the data obtained we find it consistent to conclude that the noise emitted by the gear set was due chiefly to vibration of the different plates and other surface parts of the casing. These appear to be excited by vibration transmitted through the structure of the unit from such fundamental sources as the transmission gears, pumps, oil stream and miscellaneous parts in motion causing these surface parts to vibrate at frequencies determined chiefly by their own dimensions and mechanical properties. This conclusion was checked by listening to the vibrations in selected surface plates and other parts, by means of electric conversion of the vibration and a telephone, as well as by visual observations of the wave forms by a cathode-ray oscilloscope.

BASIS OF MEASUREMENTS.

All data is given in terms of sound intensity and is in the standard scale of decibels above a sound energy flux of 10^{-16} watt per square centimeter. Although for purposes of detailed examina-



MEASURING INSTRUMENTS ON BENCH—FROM LEFT TO RIGHT, MICROPHONE AMPLIFIER, FREQUENCY ANALYZER, ACOUSTIMETER, CATHODE RAY OSCILLOGRAPH.



Close-up of Sound Insulated Test House and Companion Test Unit.

tion the microphone was moved from place to place within the enclosure, for the actual study of the sound level of the gear unit itself the microphone was so located that the readings corresponded with reasonable accuracy to the average emission of noise through an imaginary surface constituting a shell three feet from all parts of the surface of the gear set.

THE HETERODYN EFFECT.

In a considerable number of measurements, especially at intermediate speeds and sound frequencies, it was observed that the sound level fluctuated upward and downward in fairly regular cycles having periods from less than a second to as long as one minute with a decided variation from 2 decibels and up to 12 decibels. This phenomenon appeared to be caused by a periodic energy interchange between the two reduction gear units coupled together with the twisted torque shafts, or a rather complicated instance of an energy pendulum. This idea was confirmed by a series of studies with a second sound meter to measure the noise emitted by the companion gear set located outside of the test house wherein it was observed that the two units increased and decreased in sound intensity in the reverse relation to each other. In the computed data, therefore, to avoid as far as possible the probable errors due to this pendulum effect, the minimum measured values were taken for the records on the assumption that the gear units working under service load would correspond more nearly to this minimum rather than to the maximum which obviously includes the accumulations produced by the condition of heterodyning when under test.

MISCELLANEOUS.

As far as could be ascertained the foundations did not importantly modify the noise which the reduction unit would develop on any kind of mechanically satisfactory foundation and, therefore, was acceptable without requiring correction of the resulting figures.

It is also necessary to note that it was found impossible to hold the rotative speed of the coupled gear sets absolutely constant over the period of time required to take the readings, which introduced a possibility for slight deviations to the accuracy of the results.

CLOSE-UP OF SOUND INSULATED TEST HOUSE AND COMPANION TEST UNIT.

TABULATION OF READINGS.

The data obtained, carefully computed and corrected for characteristics of the microphone with relation to frequency response and ambient temperature, are tabulated herewith.

TABLE 1, PART 1.

Frequency analyses, with pumps, speed approximately 300 R.P.M. Data in decibels. Frequencies in cycles per second.

	Torque 25%	Torque 50%	Torque 75%	Torque 100%
Measurement Number.....	24	15	29	32
Measured Speeds, R.P.M.	300 to 303	305	291 to 297	299 to 304
Frequency				
Below 64	65	63	66	62
64–128	69	70	70	69
128–256	72	72	72	71
256–512	69	70	71	70
512–1024	67	66	68	67
1024–2048	61	60	60	60
2048–4096	48	49	49	49
4096–8192	35	36	36	39
All frequencies	76	76	77	76

TABLE 1, PART 2.

Same; speed approximately 500 R.P.M.

	Torque 25%	Torque 50%	Torque 75%	Torque 100%
Measurement Number.....	23	16	27	30
Measured Speeds, R.P.M.	496 to 510	489	507 to 520	500 to 506
Frequency				
Below 64	73	68	74	71
64–128	73	74	75	75
128–256	75	75	76	77
256–512	72	73	75	73
512–1024	71	72	74	71
1024–2048	64	66	65	63
2048–4096	52	54	53	52
4096–8192	39	42	40	40
All frequencies	80	80	82	81

TABLE 1, PART 3.

Same; speed approximately 700 R.P.M.

	Torque 25%	Torque 50%	Torque 75%	Torque 100%
Measurment Number.....	22	17	26	31
Measured Speeds, R.P.M.....	700 to 707	715 to 721	705 to 710	698 to 700
Frequency				
Below 64	66	68	67	70
64–128	77	77	78	78
128–256	78	78	79	79
256–512	76	76	77	76
512–1024	75	79	80	77
1024–2048	70	72	73	70
2048–4096	57	59	62	59
4096–8192	44	46	48	46
All frequencies	83	84	85	84

TABLE 1, PART 4.

Same; speed approximately 900 R.P.M.

	Torque 25%	Torque 50%	Torque 75%	Torque 100%
Measurement Number.....	21	18	25	33
Measured Speeds, R.P.M.	895 to 900	895 to 907	905 to 923	902 to 905
Frequency				
Below 64	68	69	69	68
64–128	78	80	80	79
128–256	78	80	81	80
256–512	76	76	79	76
512–1024	78	77	81	78
1024–2048	72	71	77	72
2048–4096	59	61	63	62
4096–8192	47	49	50	51
All frequencies	84	85	87	85

TABLE 1, PART 5.

Same; speed approximately 1000 R.P.M.

	Torque 25%	Torque 50%	Torque 75%	Torque 100%
Measurement Number	20	19	28	49
Measured Speeds, R.P.M.	1010 to 1055	1010 to 1020	1010 to 1030	1016 to 1017
Frequency				
Below 64	69	72	71	72
64–128	78	81	80	80
128–256	81	82	80	81
256–512	77	78	78	78
512–1024	81	80	82	81
1024–2048	77	76	78	78
2048–4096	63	64	64	64
4096–8192	50	51	51	51
All frequencies	86	87	87	87

These detailed figures are summarized below for more convenient comparison and consideration and the results are graphically illustrated by the chart included herewith.

TABLE 2.

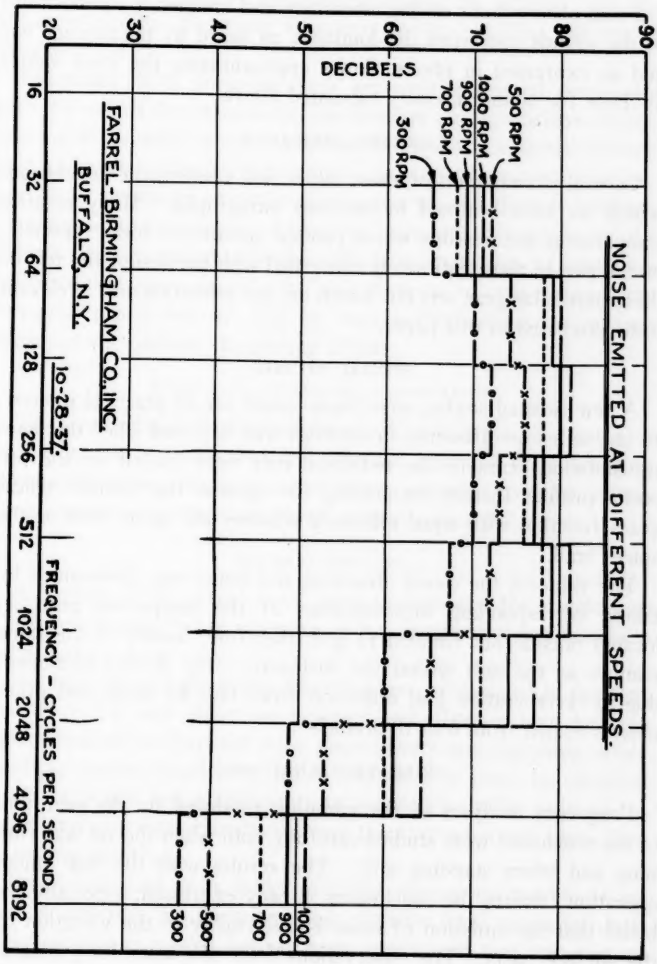
Summary of total noise, with pumps. Data in decibels.

Speed in R.P.M.	Torque 25%	Torque 50%	Torque 75%	Torque 100%
291–305	76	76	77	76
489–520	80	80	82	81
697–721	83	84	85	84
895–923	84	85	87	85
1008–1055	86	87	87	87

TABLE 3.

Average frequency analyses, all torques at each speed. Speeds are approximate. Data in decibels. Frequencies in cycles per second.

	Speed 300 R.P.M.	Speed 500 R.P.M.	Speed 700 R.P.M.	Speed 900 R.P.M.	Speed 1000 R.P.M.
Frequency					
Below 64	65	72	68	69	71
64–128	70	74	78	79	80
128–256	72	76	78	80	81
256–512	70	73	76	77	78
512–1024	67	72	78	79	81
1024–2048	60	65	71	74	77
2048–4096	49	53	60	61	64
4096–8192	37	40	46	49	51
All frequencies	76	81	84	86	87


NOISE EMITTED AT DIFFERENT SPEEDS.
DECIBELS
20
30
40
50
60
70
80
90
500 RPM
1000 RPM
900 RPM
700 RPM
300 RPM
FARREL-BIRMINGHAM CO. INC.
BUFFALO, N.Y.
10-28-37.
16
32
64
128
256
512
1024
2048
4096
8192
FREQUENCY - CYCLES PER. SECOND.
1000
900
700
500
300

LOUDNESS.

From observations of the intensities and frequency compositions of the sounds concerned the loudness, as heard by the average ear and as expressed in phons, shows approximately the same values as those for sound intensity tabulated above.

GENERAL REMARKS.

Careful examination of these tables will confirm the conclusions which we have outlined in previous paragraphs. Many detailed comparisons are possible which prompt speculative ideas especially on the part of those intimately concerned with the design and test of these particular gear sets but which are too uncertain and irrelevant to be discussed in this paper.

SPECIAL STUDIES.

A few isolated studies were made which are of practical interest. In one series no difference in readings was observed when the main transmission gears in the reduction unit were loaded on the opposite profiles thereby supporting the opinion that double helical gears function with equal efficiency whether the apices lead or the apices trail.

The value of the sound absorbing test house was determined by taking corresponding measurements of the companion gear set located outside but coupled to and, therefore, loaded in the same manner as the unit within the enclosure. The decibel increment due to reverberation and reflection from near-by walls and other objects varied from 6 to 10 decibels.

VIBRATION ANALYSES.

Frequency analyses of the vibration produced by the gear unit in the test house were studied carefully both when the set was running and when standing still. The results with the unit under operation confirm the conclusions already expressed, especially the belief that the emission of noise is due chiefly to the vibration of the surface parts. The observations from the tests for vibration characteristics of the different parts of the unit when stationary were of particular interest in view of the possibilities which they offered for similar inspection of the corresponding parts during

manufacture when changes in design can still be made. The value of such studies appeared unquestionable but the necessity for familiarity with the measuring apparatus and its handling was clearly demonstrated.

Individual parts from the gear set were examined principally with the use of the cathode-ray oscilloscope and a sensitive mallet. Bearing shells with known perfection of bond and material showed even and prolonged wave formation. As would be expected, defective shells produced irregular wave patterns and distinct evidence of dampening to a degree dependent upon the seriousness of the defects. Shafts and gear bodies with definite qualities of resonance exhibited the same smoothness and duration of wave formation. Irregular pieces such as bearing caps and miscellaneous castings displayed the obvious dampening effects.

EFFECT OF AUXILIARY EQUIPMENT.

Tests were run with various pieces of auxiliary equipment such as the pumps, both attached and detached, parts of the lubricating system and other details removed from operation and the effects of this material on the sound level carefully recorded.

GENERAL REVIEW.

A review of the actual work done to date and the conclusions which we are justified in accepting is very interesting and decidedly encouraging.

We have proved physically what has been recognized in practice, that there is a wide difference in sound and vibration level between reduction gear units of the same manufacture and operating under identical service conditions, which exposes the need for detailed study of design and manufacture to reduce all similar units to the same level and concurrently to lower the average sound level from its present position.

From full scale tests under virtually laboratory conditions we have discovered that the sound and vibration level varies but little with increase in torque but we have determined that the average intensity rises roughly in proportion to the increase in speed. Both of these phenomena check our personal observations from operating installations.

We have found it possible to isolate the behavior of different parts of the gear unit and the attached auxiliaries which should be of great help to us in the examination of a faulty installation. The delicacy of the measurements discloses the presence of low intensity harmonics and critical vibrations and it is reasonable to expect that greater familiarity with the technique involved in this work will develop further interesting and useful facts.

Outside of the information about the actual operating excellence of the reduction unit and the possibility for adjustment or replacement of annoying auxiliaries, the examinations which we have made have been of no further constructive value because they were made upon finished gear sets with which it was impossible to make any changes. Vibrating plates had to remain so and little could be done to improve conditions of resonance and gear whines unless the unit could be completely rebuilt and realigned.

"NOISE SURVEY" DURING MANUFACTURE.

It is readily apparent now that for a "noise survey" to accomplish the purpose desired it must be carried out during the process of manufacture and erection as well as at the time the finished units are on test. Defects which are discovered in the shop can be corrected at the time without undue expense or delay. The "survey" becomes an investigation and a schedule for routine examination of materials in process and of manufacturing technique. The measuring apparatus becomes an inspector's gage which obviously must require the expenditure of much time and patience to handle properly and with intelligence.

Our progress along this line is as yet unimportant but the results obtained to date are really promising. We know now that with careful examination offending plates and sources of amplification and resonance in the individual and assembled parts of the gear set housing can be located definitely and through the process of trial and error effectively corrected. It is no trouble to attach by welding dampening ribs, buttons, stars of irregular shapes or to cut holes with a torch as long as the housing has not been finish machined.

We have already used the vibration pick-up to detect defects in metal or bond in the sleeve type of bearing shell and have found this technique cruelly exact.

Already we are gaining experience with the use of the vibration meter on our gear generating machines. The results, of course, are expressed in difficulties within the gear cutting process as it progresses which nevertheless are reflected directly into errors in the gear members themselves when removed from the machine. As such their value lies in detection of faults inherent in the machine tool itself, a detail which can profitably be made to include the boring and grinding operations as well.

In the examination of the rotating gear members during the lapping and finishing operations where the results can be translated directly into inaccuracies in tooth form, pressure or helical angle, spacing and so forth, such a study actually becomes a refinement of our already established practice of determining the expected operating excellence of gear members at this stage of manufacture with the use of a flexible supporting frame and ten-thousandths dial indicators suitably placed. We know that gear whine and gear ring or resonance are detectable and capable of prediction with the use of the sound and vibration meters and we also know that great improvement in the operating characteristics of the gears themselves can be made during this lapping and finishing operation. There is every reason to expect, therefore, that the desired accuracy can be obtained and maintained when we are able to determine for a certainty on our lapping machines when this point has been reached.

POSSIBILITIES FOR IMPROVEMENT IN DESIGN.

The opportunities which these studies present in reaching the correct answer to the controversial questions of helical angle, pressure angle, tooth depth, back lash, tip relief and the necessity for lapping and finishing are quickly recognized. Noise in the lubricating system, both in the supply lines and in the drainage, can be determined and the effects of windage from the rotating members accurately measured and the possibility for whistle or siren effects discovered and corrected. The effects of the flexible or claw couplings and the causes for roughness and vibration in their operation can be studied with real thoroughness.

CONCLUDING REMARKS.

The activity and interest in this subject which has been stimulated during the past year and the programs for experimentation, examination and test which are now under way make it impossible to draw up a conclusion to this paper. We expect confidently to have made important progress during the coming twelve months in the technique of measurement and inspection of gear sets and in enlarging our experience with the handling of the sound and vibration meter equipment and to have improved to a marked degree the accuracy and operating performance of the finished reduction gear units. For the avoidance of induced vibrations and resonance from connected and related machinery we must rely as always on the thoroughness with which the entire installation has been engineered.

ONE HUNDRED YEARS OF UNITED STATES NAVAL ENGINEERING HISTORY.

By Rear Admiral Harold G. Bowen, U.S.N., Member.*

On November 3, 1937, the American Newcomen Society made its annual pilgrimage to Annapolis, Maryland, to visit a historic old town where every street and building remind us of their presence long ago at the birth-throes of a nation.

The Newcomen Society of Great Britain has for many years dedicated itself to the study of engineering history in England, and its off-shoot in this country carries on the tradition here. On both sides of the Atlantic, the rosters of membership include a most distinguished and representative group of business and professional men whose hobby it is to trace down the obscure trails of engineering development in every line.

The objective of the Society's pilgrimage this year was primarily a Naval study so that the visit was made to see the United States Naval Academy at Annapolis. During the visit, the Chief Engineer of the Navy, Admiral Bowen, delivered the principal address, discussing Naval engineering development over the last one hundred years. The subject matter is of so much interest to every member of the Society of Naval Engineers, and so provocative to all students of naval and marine engineering history, that it has been printed here in full with the kind permission of the Newcomen Society, and of the author—the prospective next President of this Society.

Mr. Toastmaster, Mr. Ambassador, Mr. Minister, Members of the Newcomen Society and Guests:

On January 23, 1837, one hundred years ago, was signed the contract for the main engines and boilers of the first steam warship to perform regular service in the United States Navy; the *Fulton* the Second.

* Chief of the Bureau of Engineering, and Engineer-in-Chief, U. S Navy.

In 1814, it is true, a remarkable steam vessel, the *Demologos*, or *Fulton* as she was afterwards named, had been built at New York under the design and supervision of Robert Fulton.

The *Demologos* was an interesting vessel; length 156 feet, breadth 56 feet, depth 20 feet, tonnage 2475, water wheel 16 feet in diameter, and 5 foot stroke; boiler, length 22 feet, breadth, 12 feet, depth, 8 feet. This vessel, hull by Adam and Noah Brown, engines and boilers by Robert Fulton, cost $320,217. The water wheel was installed in the middle of the vessel so that she had in effect two keels.

The effect of the construction of this vessel on contemporary European thought may be illustrated perhaps by an extract from a treatise on steam vessels published in Scotland at that time and purporting to describe the *Demologos*.

"Length of deck 300 feet, breadth 200 feet, thickness of her sides, 13 feet of alternate oak plank and cork wood; carries 44 guns, 4 of which are 100 pounders; quarter deck and forecastle guns 44 pounders, and further to annoy an enemy attempting to board, can discharge 100 gallons of boiling water in a minute, and by mechanism brandishes 300 cutlasses with the utmost regularity over her gunwhales; works also an equal number of heavy iron pikes of great length, darting them from her sides with prodigious force, and withdrawing them every quarter of a minute." *

The end of the War of 1812 brought the career of this vessel to an abrupt end. She was assigned to the Brooklyn Navy Yard as a receiving ship and many years afterward was destroyed by a magazine explosion with nothing but a few trial trips to her credit. This *Demologos* was the first steam war vessel built for any Navy. The Navy insisted that she be equipped with sails, a feature not contemplated by her designer. This was done, however, and marks the first official encounter between sail and steam. The sailing master won round one handsomely.

What a pity that for the next twenty years nothing was done to further the progress of steam in war vessels!

But to return to the *Fulton* the Second.

In 1835 we find the Secretary of the Navy, Mr. Mahlon Dickerson, a civilian mind you, addressing a letter to the Board of Navy

* The Steam Navy of the United States—F. M. Bennett.

Commissioners, reminding them that Congress in 1816 had authorized the construction of a steam war vessel and requesting them to proceed with the construction.

Now in 1835 steam navigation was well under way. It has been stated that as early as 1814 there were 5 steamers operating on the Thames, and on most of the rivers of North America. The *Savannah* made her historic crossing of the Atlantic in 1819. Later the *Enterprise* made a voyage from London to Calcutta, steaming 64 days and sailing 39. Engine building was similarly well advanced. You will recall that the engines and boilers of the *Clermont* were purchased by Fulton from Boulton and Watt. In 1814 we find Fulton making his own engines and boilers.

Well, the Navy Commissioners got busy, hired a Mr. Charles H. Haswell of New York, an engineer of established reputation, to get out the engine and boiler drawings, for which he was to receive the magnificent remuneration of $250.00. Later in the year Mr. Haswell was appointed Chief Engineer of this *Fulton* the Second. and so became the first Engineer Officer in the Navy.

Our first steam war vessel to actually see service was a vessel of 1200 tons displacement. The engines were built by the West Point Foundry Association of New York and consisted of two horizontal jet condensing engines located on the spar deck. The diameter of the cylinder was 50 inches and the stroke 9 feet. Each engine was coupled to a side wheel 22 feet 9 inches diameter and 11 feet 6 inches wide and the cranks and shafts were made of cast iron. The boilers were built by a private contractor at the Brooklyn Navy Yard from copper plates and rivet rods left over from the 1816 building program. For this work the contractor charged 8½ cents a pound. The original boiler installation consisted of 4 wagon shaped boilers of the return flue type, 16 feet long, 10½ feet wide and 9¼ feet high which were located in the hold under the engines. Each boiler had its own smoke pipe.

The engines developed 625 horsepower and the machinery weight dry was ¼ ton per horsepower against 1/12 ton in 1898 and less than 1/75 ton today. Her usual speed was 12 knots at 20 R.P.M. A typical report shows her carrying 6 to 11 inches of steam and 10 to 12 inches of vacuum. In those days steam gages were constructed in the smithy by bending a half-inch diameter tube into

two legs, one leg fifteen inches long and the other about thirty. The "U" was partially filled with mercury, in one column of which a light pine rod was floated. The other column was connected to the boiler and the rise and fall of the rod measured against a graduated plate served to indicate the boiler pressure.

In 1838 we find the *Fulton* ranging up alongside the British transatlantic steamer *Great Western* and passing her rapidly just outside of the Port of New York. She was employed in active service along the Atlantic Coast until 1842 when she was laid up. Later, after having had entirely new machinery installed she was assigned to active duty in 1851. The reconditioned *Fulton* made many successful cruises, was in ordinary at the Pensacola Navy Yard in 1861 when that Yard was captured by the Confederates and was burned by them in 1862 when they abandoned the Yard. Thus ended the career of the *Fulton* the Second.

1839 saw the construction of two large side-wheel frigates, the *Mississippi* and the *Missouri*. Except for the engines they were exactly alike. It is of interest to note that, in designing the boilers, they were laid down to full scale, and the dimensions determined for each plate. This marks the first time in the history of boiler manufacture that plates were rolled and trimmed to measure. The *Mississippi* also had a closed fireroom with forced draft, the origignal system having been installed in the S. S. *Rainbow* a few years before.

The *Mississippi* had two side lever engines, cylinder diameter 75 inches, stroke 7 feet; the *Missouri*, two inclined direct acting engines, cylinder diameter 62½ inches, stroke 10 feet. The cylinder volumes were almost identical, the object being to compare the merits of long and short stroke engines. As indicating construction conditions in those days we find that the *Mississippi* machinery was built by Merrick and Towne at Philadelphia, her wooden hull at the Navy Yard, Philadelphia; the *Missouri* machinery was built by the West Point Foundry Association, Cold Spring, New York, her wooden hull at the Brooklyn Navy Yard. The designs were by Principal Engineer, U.S.N., Mr. Charles W. Copeland, who was superintending engineer of the West Point Foundry Association at the time that concern was building the engines for the *Fulton*. In the meantime, costs had gone up to approximately $570,000. Both of the vessels were bark rigged, spreading 19,000

square feet of canvas. Steam is still auxiliary to sail, and very much so. This was no peculiarity of us Americans for at that time we find Lord Napier in a speech in the House of Commons declaring:

"Mr. Speaker, when we enter Her Majesty's Naval Service and face the chances of war, we go prepared to be hacked in pieces by cutlasses, to be riddled with bullets, or to be blown to bits by shot and shell; but Mr. Speaker, we do not go prepared to be boiled alive." *

The *Mississippi* had a wonderful record in service. She was the flagship of Commodore Matthew Calbraith Perry in the Mexican War and also on his expedition to open Japan to commerce. She carried Kossuth, an exile from Turkey to France, was the flagship of Flag Officer Josiah Tattnal in 1859 in the Pei Ho River, when Tattnal declared that "blood is thicker than water," and was finally lost in the joint operations of Grant and Farragut in the Mississippi River which resulted in the capture of Vicksburg.

One of the weak points in the early steam frigates was the presence of the paddles, exposed as they were to easy damage by almost any agency.

Colonel John Stevens of New Jersey, F. P. Smith of England, and Captain John Ericsson were pushing the screw propeller early in the 19th century. Smith and Ericsson patented propulsion by a propeller in London in 1836.

Captain Robert F. Stockton, U.S.N., later of fame in the acquisition of California, became interested while in England with Ericsson's work and persuaded him to come to this country. Stockton, with Navy Department backing, purchased two iron boats of Messrs. Lairds of Birkenhead which he caused to be equipped with engines and screw propellers of Ericsson design.

As a result of Stockton's agitation, the *Princeton,* named after Stockton's home town in New Jersey, was laid down in 1842 at the Philadelphia Navy Yard. Merrick and Towne constructed the machinery but to Ericsson's design.

The propeller also designed by Ericsson had a cast brass hub with six arms, which were surrounded by a copper annulus to which six brass blades were secured by riveting, the arms and blades presenting a true helicoidal surface.

* The Steam Navy of the United States—F. M. Bennett.

In the *Princeton's* engines a rectangular piston vibrated like a pendulum. Novel as this sounds, the principle already had been presented by Watt. Ericsson was, however, the first engineer to make a go of it. Opposite the semi cylinder was a smaller one with its piston merely a prolongation of the main piston. Both were acted upon by the steam at the same time, the small cylinder merely producing negative work. Two of these engines were installed parallel to the crankshaft which they actuated.

The *Princeton* scored a number of firsts. She was the first war steamer ever fitted with a screw propeller although H.M.S. *Rattler* was a close second; the first man-of-war with all her machinery below the water line, and so protected against gunfire; the first man-of-war designed to burn anthracite; the first to have a telescopic smoke pipe; and the first to have her engine directly connected to the propeller shaft. She logged 14 knots on her trials, and her mean steam pressure was in excess of 11 pounds.

In 1845 the British Admiralty demonstrated the superiority of the screw propeller over the paddle wheel in a most forthright manner. The *Rattler* and the *Alecto* were secured stern to stern. Both vessels were almost identical except the *Rattler* was screw propelled and the *Alecto* paddle propelled. With the engines of both ships operating at full power, the *Rattler* towed the *Alecto* astern as a speed of 2½ knots.

One of the curious features of the times was the fact that the Merchant Marine was very slow to take up the screw propeller compared with the British and American Navies.

The Encyclopedia Britannica tells this story on itself: "Fincham writing in 1851, says that in England engineers were reluctant to admit the success of the screw propeller and adds: 'A striking instance of prevailing disinclination to the screw propeller was shown on the issue of a new edition of the Encyclopedia Britannica in which the article of steam navigation contained no notice whatever of the subject.'"

The services of the *Princeton* were eminently satisfactory and it is said that her successful performance did much to further the progress of steam.

In 1844 Henry Worthington patented the steam pump, though not applied to Naval use until 1850 in the *Powhatan*. Before this innovation all pumps were driven off the main engine.

1847 saw the first employment of zinc plates in boilers to reduce the effects of corrosion.

In 1846 Ericsson applied the surface condenser to a revenue cutter. This was later developed by Chief Engineer Sewell of the Navy into something resembling its present form. Until 1846, salt water was used in the Navy for boiler feed. Corrosion did not become a serious problem until the advent of steel boilers and high steam pressures. 1848 likewise marked the introduction of the steam steering engine.

At this period indicators and counters were unknown, horsepower was estimated from coal consumption.

After the Mexican War several large steam warships were built. Engine frames were introduced. They were the box girder type of wrought iron. The donkey or port use boiler also made its appearance—particularly for supplying steam to the coal hoisting engine. Previously it had been necessary to stop the main engines in order to have steam enough to hoist ashes.

Steam was in the ascendant, but not without its enemies. 1851 finds the Secretary of the Navy recommending the construction of two vessels, one of sail and one of steam. He also recommended the construction of steam engines in Navy Yards, thereby anticipating in part the character of present legislation.

In 1853 Secretary Dobbin is urging Congress, with success, to construct steam vessels with screw propellers. He states "Steam is unquestionably the great agent to be used on the ocean, as well for purposes of war as for commerce." These steam vessels were essentially frigates, the steam power being auxiliary in spite of the fact that many advocated "that new constructions should be provided with full steam power, with sails as an auxiliary."

At this period we find our latest steam frigates capable of almost 9 knots. The propeller was turning over at 46 R.P.M. Steam pressure was still low, 13.5 pounds per square inch. The jet condensers were good for 24.5 inches of mercury. Total horsepower had risen to 1300 and it took 3½ pounds of coal to produce one horsepower. The Stephenson link and all kinds of patented cut-offs are very much in evidence. A rudimentary feed heater has appeared in connection with the boiler. The popular type of boiler is the Martins 4 furnace vertical water tube fabricated of iron ex-

cept that the tubes were of brass. All smoke pipes were telescopic so that they could be lowered in port and not spoil the appearance of the ship.

The Civil War gave a tremendous impetus to steam.

In 1861 Congress authorized the construction of 14 steam sloops. The Navy Department, on its own responsibility, ordered 23 small, heavily armed, gun boats. Many double ended river boats were also constructed. All of these types, of course, for blockade or river work. Paddle wheels disappeared except for shallow draft vessels. In 1861 there were built or authorized 49 steamers and the Secretary of the Navy stated "No sailing vessels have been ordered to be built, for steam as well as heavy ordnance has become an indispensable element of the most efficient Naval power."

After 1861, the sail exists only as an auxiliary to steam. In fact, the new *Ironsides,* a wooden vessel whose sides were sheathed with 4 inches of iron plate in the wake of her main battery, had her masts removed upon reporting for duty at Port Royal.

The *Monitor* appears in 1862. Every part of her was designed by John Ericsson and she represented what a great engineer, who was not a sailing master and who was not tied down by tradition, thought a warship of the period should be. She had the usual Ericsson vibrating type engine, screw propeller, and no sails. The contract required the contractors to "furnish masts, spares, sails, and rigging of sufficient dimensions to drive the vessel at the rate of six knots per hour in a fair breeze of wind." This clause was ignored by Ericsson and apparently forgotten by the Navy Department. The contract also required that if the vessel failed in speed or other particulars, all moneys paid should be refunded in full to the United States. She failed to meet the contract in speed and also in some other particulars but fortunately for the contractors and for the Union, the *Monitor* met the supreme test of battle with the *Merrimac* and the contracts were paid in full within a week after that historic encounter. Credit for proceeding with "Ericsson's Folly" is due to a great Secretary of the Navy, Gideon B. Welles, and to Ericsson's backers, Griswold, Winslow, and especially C. S. Bushnell.

The London Times commented as follows upon the battle between the *Monitor* and the *Merrimac*—"Whereas we had available

for immediate purposes 149 first-class warships, we have now two, these two being the *Warrior* and her sister *Ironside*. There is not now a ship in the English Navy, apart from these two, that it would not be madness to trust to an engagement with that little *Monitor*." *

The story of the *Monitor* must always amaze the engineer. Here was a completely revolutionary piece of machinery, built directly from drawings, passing through no period of experiment and development, and succeeding wonderfully well under such circumstances. It is true that at the time her importance may have been somewhat exaggerated but the battle between the *Monitor* and the *Merrimac* enormously influenced warship construction and resulted in our country, for a time at least, approximating European developments in ironclads.

We built many more monitors. An energetic and able Assistant Secretary of the Navy, G. V. Fox, vigorously pushed the building program. Ships were getting more complex and consequently more expensive. A big monitor cost $1,150,000. The *Dunderberg* completed after the end of the Civil War, repurchased by the contractor and sold to France was a wonderful vessel for her day, 7000 tons displacement, 5000 horsepower. She was an ironclad resembling the *Merrimac* rather than the *Monitor* and was designed with double bottoms, collision bulkheads and a system of transverse water-tight longitudinal bulkheads. The machinery was enclosed in water-tight bulkheads. Independent air and circulating pumps as well as bilge pumps added to her handiness. She made 80 R.P.M., or 15 knots at full power. Her shaft bearings were cored for water circulation. As the *Rochambeau* she was regarded as an exceptionally formidable addition to the French Navy. The machinery of such a ship cost about a half a million.

When Chief Engineer Isherwood designed the machinery of the *Wanpanoag* and *Ammonoosuc* in 1863, the Navy Department evidently had in mind the raiding of British commerce in the event of European interference in our Civil War. This was perfectly natural in view of the memories of the damage inflicted on Union commerce by the Confederate cruisers, *Florida* and *Alabama*. The *Wampanoag*

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made 17 knots on trials, 2 or 3 knots more than the fastest vessels then in existence. She had 8 vertical tubular boilers and 4 superheater boilers and step-up gears with wooden teeth to increase the speed of her propeller. Naval opinion was decidedly against these vessels when they ran their trials in 1868 and they were speedily retired. It was 10 years before her speed was equalled in the British Navy and 21 years before it was equalled in the Navy of the United States. Perhaps the fault with the *Wampanoag* and her sister ship was that they were too far ahead of their time. At any rate, they illustrated the great ability of Isherwood as a designer.

Although simple horizontal engines had been installed in the *Wampanoag,* double expansion engines began to appear in 1871. An increase in steam pressure from 30 to 80 pounds, necessary to supply the new engines was responsible for changing the boilers from the rectangular to the familiar cylindrical form of the Scotch boiler. From then on changes in steam machinery occur very rapidly.

Although during the period subsequent to the Civil War we find Naval steam shipbuilding at a low ebb, considerable engineering experiment took place.

In 1867 we find a Navy Board experimenting with the combustion of petroleum in boilers, and furthermore with the atomization of fuel oil.

Twelve years later mineral oil replaced vegetable and animal oil as a lubricant, and that same year Chief Engineer Baird brought forth a drowned tube circular coil evaporator which relieved the boiler of its former dual functions. The introduction of the evaporator opened the way for feed water control and made possible the use of steel boilers and high pressures, temperatures and steaming rates.

As I have said before, the Civil War gave a decided impetus to steam. After the Civil War an equally decided reaction set in. In 1869 in Navy Department General Orders we are astonished to find: "Hereafter all vessels of the Navy will be fitted with full sail power. The exception to this will be the tugs and dispatch vessels not fitted with sails."*

* The Steam Navy of the United States—F. M. Bennett.

A following order enjoined rigid economy in the use of coal, stated regulations would soon be issued determining the number of times it would be permitted to get up steam to turn over the main engines, and finally warning commanding officers that they themselves would have to pay for any unusual expenditure of fuel. The promised regulations were never issued due to a change in Secretaries of the Navy.

Even in 1871 the annual report of the Navy Department refers to steamers as "sailing vessels with auxiliary screws."

The dispute between the proponents of 2- and 4-bladed propellers was finally settled in 1875 in favor of 4 blades. A ship sailed better with 2 blades and steamed better with 4 blades. Steam won!

While the controversy between the 2-bladed and 4-bladed propeller was in progress Isherwood was conducting the first practical tests on propellers at the Navy Yard, Mare Island. The results of these tests formed the basis of propeller design for 27 years thereafter until the advent of Chief Constructor Taylor's treatise on propellers. Shortly afterwards, Rear Admiral Dyson continued the work of Isherwood and Taylor and Dyson's work is standard today. Similar work was in progress during this period by Froude in England.

Congress appears to have had no illusions about what was going on in the world and in an Act of 1873 refers to "steam vessels of war with auxiliary sail power."

By 1881 it became apparent that a complete rebuilding program for the Navy had become necessary. In 1882 Congress authorized the first ships of the famous white squadron—the *Atlanta, Boston, Chicago,* and *Dolphin,* all equipped with auxiliary sail.

1885 saw the introduction of the triple expansion engine in the *Newark* and *Yorktown.* Steam pressure rose to 160 pounds per square inch.

The battelship *Maine* authorized in 1886 marked the appearance of the vertical, triple expansion engine due to the activities of Melville who became Engineer-in-Chief in 1887. Melville was especially picked by Secretary of the Navy W. C. Whitney for the position on account of his wonderful record with the ill-fated Jeannette Polar Expedition.

The armored cruiser *New York* completed in 1893 cost about $3,000,000 and was built by Wm. Cramp & Sons. She had two large vertical triple expansion engines per shaft, twin screw, and logged 21 knots on trial. Life is beginning to get complicated for the engineer. The *New York* had, including auxiliaries, 92 steam engines with 170 cylinders, not to mention the electrical installation which was growing with every ship laid down.

The standardization of vessels over the measured mile was introduced by Melville.

In 1893 the *Columbia* and *Minneapolis* appear with triple screws and sustained speed approximating the best of the transatlantic liners, and in the same year a Ward water tube coil boiler was installed in the *Monterey*.

The Yarrow water tube boiler with 250 pounds per square inch pressure was installed on the *Nashville* in 1895 and in the same year a Babcock and Wilcox boiler improved by William Hoxie was installed in the *Alert*.

In 1897 Sir Charles Parsons startled the world with his Turbinia making 32.61 knots on trial. This was rapidly followed in England by two successful turbine driven destroyers, the *Viper* and the *Cobra* and the turbine cruiser, the *Amethyst* and the first turbine battleship, the *Dreadnought*, 1906.

Under Melville, our battleship speed was increased from 16-18 knots.

The Curtis turbine was first applied to marine use, I believe, in 1902 in the yacht *Revolution*. In 1904 a 5000 H.P. Curtis installation was made in a power plant in Chicago. The growth of the large electrical generating stations ashore began to influence marine engineering and this influence has continued to an increasing extent ever since.

In 1904 the *Birmingham* was equipped with triple expansion engines, the *Chester* with Parsons turbines, and the *Salem* with Curtis turbines. These cruisers were identical for all practical purposes. Their speeds on trial were as follows:

Chester, Parsons turbines	26.52
Salem, Curtis turbines	25.94
Birmingham, reciprocating engine	24.32

The *Birmingham* had much the better economy, she was the more reliable ship, but the remarkable thing this comparison reveals is the incredible progress of the steam turbine against its well established rival. All of these turbines were direct connected to the propeller shaft and, therefore, could not be operated at the high speeds necessary for a turbine to develop its highest economy.

Comparative tests were available later, upon completion of the battleships, *North Dakota,* Curtis turbine, and *Delaware,* a reciprocating engine. The *North Dakota* used 43 per cent more fuel at 12 knots than the *Delaware.* In spite of this other battleships were built with turbines until 1910 when the *New York, Texas,* and *Oklahoma,* the last of the reciprocators were authorized and they are still in service. They were equipped with forced lubrication, an improvement which came in with the U. S. S. *Delaware.*

The propeller is essentially a low speed mechanism, and the turbine a high speed mechanism. Furthermore, a Naval vessel does a high percentage of its cruising at low speeds.

The desirability, if not the necessity, for reduction gearing between the turbine and the propeller shaft, if the turbine were to be developed to its highest economy in marine engineering soon became apparent.

The reduction gear appeared in the *Pennsylvania, Nevada, Arizona,* and *Idaho* for cruising turbines only, which were unclutched at high speeds. The *Arizona* had Melville-MacAlpine reduction gears with a floating hydraulic frame. All other types were rigid; the herring bone helical gear was universal.

The unsatisfactory economy of the direct-connected turbine soon lead to two distinct developments, the electric drive, which is nothing but an electric reduction gear, and the geared drive.

Electric drive had been mentioned as early as 1907 by Dr. Emmett in the Scientific American.

The Bureau of Engineering decided to take a look at electric drive. Under Engineer-in-Chief, H. I. Cone, the *Neptune,* a collier, was equipped with Westinghouse reduction gears, the *Maumee,* a tanker, with M. A. N. Diesel engines manufactured at the Brooklyn Navy Yard and the *Jupiter,* another collier, with General Electric electric drive. The *Maumee* installation never proved entirely satisfactory and except for Diesel engines in submarines and small

craft, this type of drive has not been used for other Naval vessels during the period covered by this paper. The results on the *Jupiter*, now known as the aircraft tender *Langley*, were so successful and so satisfactory from a Naval point of view that electric drive, both by General Electric and Westinghouse was installed in 6 battleships and two large aircraft carriers. Great credit for the development of the electric drive is also due to Rear Admiral H. I. Cone, and the late Rear Admiral R. S. Griffin, Chiefs of the Bureau of Engineering, respectively, from 1909 to 1913, and 1913 to 1921, Admiral S. S. Robison, Commanding Officer of the *Jupiter* in 1914, and Rear Admiral S. M. Robinson, first engineer officer of the *Jupiter*.

During the World War large numbers of destroyers were built with reduction gears, some of which gears were satisfactory and some were not. The laminated gear looked attractive but didn't work out in practice.

Consequently, when 10,000-ton cruiser building started in 1924 a great impetus was given to gear building which resulted in the installation of reduction gears in all of our 10,000-ton cruiser program.

When the question of the modernization of the electric drive battleship *New Mexico* arose in 1929 the necessity for installing more horsepower in the same space absolutely precluded electric drive on the basis of weight, space, and cost, and geared drive was selected.

The electric drive is very attractive from a Naval point of view and it is to be hoped that new improvements and inventions will cause it to become again a lively competitor of the reduction gear.

I have tried to present the high lights of the story of steam power in our Navy; the 100 years struggle between steam and sail for supremacy. Time and space have required me to omit much which I would like to have presented. I have scarcely touched the phenomenal advance of electricity, the development of the steel vessel and the evolution in armor and armament—all intimately connected with the story of steam.

The advent of extremely high pressures and temperatures in power plants, together with the development of forced circulation boilers in this country and Europe, will unquestionably influence marine and especially Naval design for some time to come.

A careful and studied approach must precede the adaptation of power plant pressures and temperatures to ship's use, but with the high premium placed on space, weight, and economical performance in Naval machinery, power plant design will continue to exert a compelling pressure on Naval engineering designs.

BIBLIOGRAPHY.

Bennett, Steam Navy of the United States.

Admiral Preble, History of Steam Navigation, Encyclopedia Britannica.

Secretary of the Navy Reports, Annual.

JOURNAL, AMERICAN SOCIETY OF NAVAL ENGINEERS (various).

Proceedings, U. S. Naval Institute (various).

Transactions, American Society of Mechanical Engineers.

Ship's Data Book, U. S. Navy.

THE DESIGN OF FUEL OIL PIPING FOR MARINE PURPOSES.

By JOHN A. RAIDABAUGH, M. E., Member.*

In this carefully prepared, and well-arranged article, Mr. Raidabaugh has compiled an exhaustive resume of the contemporary design data available—frequently conflicting—which bears on this important detail of marine engineering design. After averaging, and weighing the data, he has evolved simple formulae for its application in marine drafting rooms so that further reference to the authorities becomes unnecessary. In order that the subject may be completely covered in one place, and without additional reference, tables have been included, appropriately acknowledged in the bibliography, for which use due acknowledgment is made. The article is recommended for study by piping specialists and marine engineering designers. It is too technical to interest the casual reader.

INTRODUCTION.

Due to difficulties which have arisen in service, particularly in pumping fuel oils at high viscosities, a survey has been made of available data with the purpose of preparing standard design data on fuel oil piping for Marine service. This survey has been made under the direction of the Naval Boiler Laboratory, Navy Yard, Philadelphia, and it has been necessary to bear in mind that fuel oil piping design for ship work differs from commercial design in several respects.

(*a*) In pumping oil aboard ship the limits are mainly on the suction side of the pump, while in commercial pumping the discharge pressure is the important factor.

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(*b*) In designing piping for ship work, it is, of course, necessary to make the pipes large enough for the service intended, but the design must be fairly accurate as a factor of safety cannot be added without unduly increasing the weight of piping and the space required, both of which are of prime importance in ship design.

(*c*) Pressure drop due to valves and fittings is of much more importance in ship design as the pipe lines are comparatively short in comparison to the number of valves and fittings necessary.

(*d*) For ship work, especially Naval vessels, the range of oils and piping used is not as large as for ordinary commercial design.

For Naval vessels, certain details of piping design, such as grade of oil used, viscosity at which the oil is to be pumped, pressures of the various systems, pipe materials and standard pipe dimensions are usually specified. As such data is also available for merchant vessels, the design problem resolves itself mainly into the selection of the pipe sizes which will meet the allowable velocities and pressure drops in the various systems and under the various conditions.

The pressure drop is made up of two factors:

(*a*) Pressure drop due to static head or lift.

(*b*) Pressure drop due to friction of oil in pipes.

As the pressure drop due to static head or lift is readily obtained from the location of the pump in relation to the suction or discharge level, the problem finally resolves into an investigation of friction loss and of an orderly procedure for combining this loss with the other known or specified factors.

THE SURVEY.

The information and conclusions in this article are based upon a study of many text-books, hand-books, pamphlets and technical articles and upon the known requirements for Naval and Merchant vessels.

1. All of the references studied agree that stream line flow occurs at low velocities, and that as the velocity is increased, some point is reached where the type of motion suddenly changes from stream line flow to turbulent flow. The point at which this change takes place is the critical velocity and above this critical velocity there is a critical range in which the flow may be either stream line or turbulent.

It is further agreed that in stream line flow the friction loss follows a definite mathematical law, but that in turbulent flow it is necessary to use a friction factor based upon experimental data. This friction factor (f) is a function of an expression known as Reynolds Number or Reynolds Criterion (R). The relation between "f" and "R" is definite for stream line flow but is based on experimental data for turbulent flow. The Reynolds number is also used to locate the critical range, which should be avoided in design as flow in this critical range produces excessive pressure drops without corresponding increases in volume discharged.

Stream line flow is variously described but Nelson [1] gives the following definition: "Through a circular conduit, the fluid travels as a series of cylinders which slip past one another. Flow at the center of the pipe progresses at the greatest velocity and the cylindrical film at the wall is almost stagnant."

Turbulent flow is variously described, but McAdams [2] says that it "is characterized by the presence of innumerable eddy currents in the stream and with little or no motion at the pipe walls."

2. The "Friction Pressure Drop" formulas studied are given in various nomenclatures; some use c.g.s. units and some use English units; some are based on velocity and some on quantity of oil; some give pressure drop in pounds per square inch, some in pounds per square foot and some in feet of head.

From a comparison of these formulas it is found that they can all be resolved back to the same basic formulas, which are as given below.

TURBULENT FLOW.

$$\text{(I) Pressure Drop} = \frac{\text{Constant} \times \text{Friction Factor} \times \text{Length} \times \text{Specific Grav.} \times (\text{Velocity})^2}{\text{Inside Diameter}}$$

in which the friction factor is a function of the Reynolds number or criterion (R) according to the units used and

$$\text{(II) R} = \text{Reynolds Number} = \frac{\text{Inside Diam. (Feet)} \times \text{Velocity} \times \text{Density}}{\text{Absolute Viscosity (Poundals)}}$$

or

$$\text{(III) R} = \text{Reynolds Criterion} = \frac{\text{Inside Diam. (Inch)} \times \text{Velocity} \times \text{Specific Grav.}}{\text{Absolute Viscosity (Centipoises)}}$$

[1] Reference figures refer to "List of References" given at end of article.

STREAM LINE FLOW.

In stream line flow the friction factor is a definite function of "R" at all points and if this definite value is substituted in formula (I) we have

$$\text{(IV) Pressure Drop} = \frac{\text{Constant} \times \text{Absolute Viscosity} \times \text{Length} \times \text{Velocity}}{(\text{Inside Diameter})^2}$$

In both of these pressure drop formulas (I) and (IV), the constant is dependent upon the units used.

3. Before selecting any proposed formula or units to be used for standard design data, a study was made of the data available for the various factors which are necessary for the calculation of friction pressure drop. These factors may be divided into three classes as follows:

Basic Information:

(*a*) Basic oils to be used.
(*b*) Viscosity at which oils are to be pumped.
(*c*) Viscosimeter scales.
(*d*) Conversion between viscosimeter scales.
(*e*) Effect of temperature on viscosity.
(*f*) Effect of pressure on viscosity.
(*g*) Effect of heating and cooling of oils.

Internal Factors:

(*h*) Absolute viscosity.
(*i*) Kinematic viscosity (ratio between absolute viscosity and specific gravity).
(*j*) Specific gravity or density.
(*k*) Conversion of various gravities.
(*l*) Effect of temperature on specific gravity.

External Factors:

(*m*) Quantity of oil flowing.
(*n*) Inside diameter of pipes.
(*o*) Velocity of flow, which can be obtained from items (*m*) and (*n*).
(*p*) Length of pipe.

(*q*) Friction loss through valves and fittings.
(*r*) Friction factor and critical range.
(*s*) Type, size and roughness of pipe.

4. *Basic Oils to be Used (Item 3a):* The basic oils for use on Naval vessels are as follows:

(*a*) Navy grade fuel oil—Viscosity 65 S.S.F. at 122 degrees F. (Maximum)—A.P.I. gravity of 8 degrees (Minimum).
(*b*) Bunker "C" fuel oil—Viscosity 300 S.S.F. at 122 degrees F. (Maximum)—A.P.I. gravity of 2 degrees (Minimum).
(*c*) Diesel Fuel Oil—Viscosity about 45 S.S.U. at 100 degrees F.—A.P.I. gravity of 25 degrees (Minimum).

While the above requirements may be changed for merchant vessels they represent good practice and are recommended for general use.

5. *Pumping Viscosities (Item 3b):* The basic data recommended for marine service is as follows:

(*a*) The capacity requirements of the following systems should be met while pumping fuel oil at a viscosity of 700 S.S.F., which is equivalent to a pumping temperature of about 65 degrees F. for Navy grade and 105 degrees F. for Bunker "C".

Booster and Transfer pump suction and discharge for general transfer service and for discharge to side of ship.
Booster and Transfer pump discharge to service pump suction.
Service pump suction connections to service tanks.
Service pump discharge to fuel oil heater inlet.

(*b*) Viscosity of fuel oil in discharge from heater to boiler front manifold should be about 150 S.S.U. and for purposes of uniformity this will be considered as 20 S.S.F. which is equivalent to a pumping temperature of about 180 degrees F. for Navy grade and about 225 degrees for Bunker "C".

(*c*) Piping between fuel oil tank drain pumps, separating tanks, oil and water separators, etc., should be designed for the same pumping conditions as booster and transfer system.

(*d*) Usually there are no heating coils in the Diesel fuel oil storage tanks and this oil must be pumped at the temperature of the oil in the tanks, which temperature is assumed to be 50 degrees F. This is equivalent to about 70 S.S.U. for present "Navy" Diesel fuel oils.

6. *Viscosity (Viscosimeter Scales) (Items 3c and 3d):* Viscosity is variously defined but Nelson [1] says "The viscosity of an oil is a measure of its resistance to internal flow and is an indication of its oiliness in lubrication of surfaces." The actual unit of viscosity is *absolute viscosity,* and although common viscosimeters do not read in absolute viscosity, they do employ arbitrary scales, which can be converted from one to the other and also to absolute and kinematic viscosity. As will be shown later, the kinematic viscosity is the most convenient unit for use in calculations.

There are many of these arbitrary viscosity scales such as Engler, Redwood, Redwood Admirality, Barby, Saybolt Universal, and Saybolt Furol. There are two general types of conversion charts, one being in the form of curves and the other in the form of parallel lines. From the basic oils given above it is evident that for purposes of marine design it is only necessary to consider Saybolt Universal (S.S.U.) and Saybolt Furol (S.S.F.) and a parallel line conversion chart may be incorporated with kinematic viscosity charts, as described later.

7. *Effect of Temperature on Viscosity (Item 3e):* Each oil has a definite viscosity at a given temperature and this viscosity varies with temperature, decreasing as temperature increases. This is of importance as we have given, a basic oil and the viscosity at which it is to be pumped and in order to obtain the proper specific gravity or density of oil being pumped it is necessary to know the pumping temperature. For example, Navy grade fuel oil has a viscosity of 65 S.S.F. at 122 degrees F. and certain lines must be designed to handle this oil at 700 S.S.F. In order to obtain the correct specific gravity it is necessary to obtain the temperature corresponding to 700 S.S.F.

There are a number of conversion charts given, most of which are not entirely satisfactory for various reasons. Some charts only pertain to certain oils, some require that the base of the oil be known, some require that the viscosity be known at a certain temperature and others, particularly the A.S.T.M. Method D341-2T, requires that the viscosity be known at two temperatures.

A very convenient conversion chart for purposes of design data is the Viscosity-Temperature Diagram,[3] which covers S.S.F., S.S.U. and Engler from 10 degrees F. to 500 degrees F. and which states that it is accurate within 10 degrees for lighter fuel oils up to 200

degrees F. and for heavy oils of high flash point, it is accurate for practical application up to 250 degrees F. For accuracy it would be preferable to use the A.S.T.M. chart (D341-2T) if the viscosity at two temperatures could be ascertained.

The viscosity-temperature diagram only gives the S.S.U. scale up to 500 seconds, but above this point the S.S.U. scale is approximately ten times the S.S.F. scale. This chart has combined with it a parallel line conversion between the above scales and with kinematic viscosity. From this chart, for the example given above, it is readily seen that for Navy grade oil the temperature corresponding to 700 S.S.F. is 65 degrees F.

8. *Effect of Pressure on Viscosity (Item 3f):* The effect of pressure on viscosity is not great and except for very high pressures it may be neglected. As no very high pressures are used on ship fuel lines, the pressure effect need not be considered.

9. *Effect of Heating and Cooling of Oils (Item 3g):* All of the references studied give friction drop formulas based on isothermal flow and some of them discuss the effect of heating and cooling.

From a study of the discussions given by Nelson,[1] McAdams,[4] Kinney,[5] Danforth [6-7] and Lubrication [8] it is seen that the effect of heating or cooling is very small in turbulent flow but that it is of importance in stream line flow. Heating the oil during pumping will decrease the friction factor while cooling will increase it. The condition usually met is that of a hot oil being cooled because of a lower external temperature and this may increase the friction factor and hence the friction drop from 10 to 50 per cent. However, Danforth [6] states that this allowance should be under 20 per cent for the usual installations. He further [7] gives a method of computing mean viscosity due to temperature drop in line.

In calculations for Marine installations, the stream line flow applies mainly to high viscosities at temperatures from about 65 degrees F. to 105 degrees F. As these temperatures are so near the temperature of the machinery spaces, there will be little temperature gradient and its effect may be neglected. The possible effect of cooling the discharge from heaters to boiler front manifolds (180 degrees F. to 225 degrees F.) may also be ignored as the viscosity is low and the flow will be turbulent.

10. *Absolute Viscosity (Item 3h):* As mentioned above this is the actual unit of viscosity and is used directly in formulas II and

III for values of "R" and formula IV for pressure drop in stream line flow. Nelson[1] gives the following definition: "In the c.g.s. system the unit of viscosity is the poise or centi-poise (0.01 poise). Viscosity may be defined as the force in dynes required to move a plane of one square centimeter area, at a distance of one centimeter from another plane of one square centimeter area, through a distance of one centimeter in one second."

Similarly, the unit of absolute viscosity in English units is called the poundal and the two systems may be converted as follows:

Poises $\times$ .0672 = Poundals, or

Centipoises $\times$ .000672 = Poundals.

The absolute viscosity may be calculated from the kinematic viscosity, which is the more convenient unit to use and which is absolute viscosity divided by specific gravity or density according to the units used.

The unit of viscosity usually used in engineering calculations is the centipoise = 0.01 poise, but many of the articles studied use formulas based on viscosity in poundals.

11. *Kinematic Viscosity* (*Item 31*): Kinematic viscosity is the most convenient unit of viscosity for use in calculations and is the ratio of absolute viscosity to specific gravity or

$$\text{(V) K.V.} = \frac{\text{Absolute Viscosity in c.p.}}{\text{Specific Gravity}} \text{ for c.g.s. units and}$$

$$\text{(VI) K.V.} = \frac{\text{Absolute Viscosity in poundals}}{\text{Density (pounds per cubic feet)}} \text{ for English units, in}$$

which c.p. $\times$ .000672 = Poundals and density = 62.3 $\times$ Specific Gravity.

As mentioned above, the Viscosity-Temperature Diagram[3] includes a parallel line conversion between the viscosimeter scales and kinematic viscosity. However, formulas are given for computing the kinematic viscosity:

$$\text{(VII) Saybolt Universal, K.V.} = \frac{\text{c.p.}}{\text{S. Grav.}} = .220 \times \text{Seconds} - \frac{180}{\text{Seconds}}.$$

(VIII) Saybolt Furol, K.V. $= \frac{\text{c.p.}}{\text{S. Grav.}} = 2.04 \times \text{Seconds} - \frac{160}{\text{Seconds}}$.

These formulas are satisfactory for the range of viscosities used in Marine design but do not apply to very low viscosities. For example, with S.S.U. of 28.6 seconds the kinematic viscosity would equal zero from formula (VII).

12. *Specific Gravity and Density (Items 3j and 3k):* Specific gravity is an expression of the density or weight of a unit of material, or more specifically for purposes of this article it is the ratio of weight of a unit volume of oil to the weight of the same volume of water at 60 degrees F. Some of the references studied considered that it should be expressed in units relative to water at 39 degrees F. or 4 degrees C. which is the temperature of maximum density. However, the use of 60 degrees F. seems to be well established and will be used.

A.P.I. (American Petroleum Institute) gravity and Baume gravity are also expressions of density and the following conversion formulas are given:

(IX) Degree A.P.I. $= \frac{141.5}{\text{Spec. Grav.}} - 131.5$ or Spec. Grav. $= \frac{141.5}{^{\circ}\text{A.P.I.} + 131.5}$.

(X) Degree Baume $= \frac{140}{\text{Spec. Grav.}} - 130$ or Spec. Grav. $= \frac{140}{^{\circ}\text{Baume} + 130}$.

For purposes of Marine design it will only be necessary to use formula (IX) as the fuel oils usually have their gravities given in degrees A.P.I.

Density is the weight of a unit volume and for water at 60 degrees F. is 62.3699 pounds per cubic foot or 8.328 pounds per gallon. For purposes of conversion of formula in this article it is considered as 62.3 pounds per cubic foot and

(XI) Density of Fuel Oil $= 62.3 \times$ Specific gravity.

Various conversion tables between specific gravity, A.P.I. and Baume are given but the most complete are the tables contained in Bureau of Standards No. 154.[9] Tables No. 4 and No. 5 of this standard give comparative specific gravity and weight per gallon of each .1 degree A.P.I. from 10 degrees to 100 degrees A.P.I. inclusive, but do not give figures below 10 degrees A.P.I. (or specific gravity of 1) and therefore are not satisfactory for Navy grade or Bunker "C" oil (8 degrees and 2 degrees A.P.I.). However, the A.S.T.M.[10] standards (p. 156) include an abridged table which gives this same comparative data for each 1 degree A.P.I. from 0 degree to 100 degrees A.P.I. and this table is very satisfactory for general use and may be used to avoid the necessity of conversion by use of formula (IX).

13. *Effect of Temperature on Specific Gravity (Item 3-l):* Specific gravity of an oil changes with temperature, decreasing as temperature increases. Many of the references studied give curves to show this change in specific gravity but in this case also the most complete tables are given in the Bureau of Standards No. 154.[9] This includes:

Table No. 1—Reduction of observed degrees A.P.I. to degrees A.P.I. at 60 degrees F.

Table No. 2—Volume at 60 degrees F. occupied by unit volume at indicated temperature.

Example: If at 60 degrees F. the A.P.I. gravity is 22, one gallon of this oil at 105 degrees F. will have a volume of .9822 gallons at 60 degrees F., or in other words—

(XII) Volume of oil at indicated temperature and degrees A.P.I. at 60 degrees F. $\times$ table factor = volume at 60 degrees F. or inversely

$$\text{Volume at indicated temperature} = \frac{\text{Volume at 60 degrees F.}}{\text{Table Factor}}.$$

Table No. 3—Reduction of observed specific gravity to specific gravity at 60 degrees/60 degrees F.

As stated above these tables do not go below 10 degrees A.P.I. but the A.S.T.M.[10] standards, page 314, give a set of abridged tables, to be used when extreme accuracy is not required and these

include all degrees A.P.I. from 0 degree to 100 degrees and may be used to obtain the specific gravity corrected for temperature.

14. *Quantity of Oil Flowing (Item 3m):* For Marine design the quantity of oil is generally specified in gallons per minute for transfer pumps and is assumed to refer to G.P.M. at pumping temperature. For purposes of oil to burners it is always specified in pounds per hour which can readily be converted to G.P.M. at pumping temperature, by the use of the A.S.T.M. tables mentioned above.

For any given size of pipe there is, of course, a definite relationship between quantity and velocity and formula based on Equations I and IV may be given, using quantity instead of velocity.

15. *Pipe Diameter (Item 3n):* All of the formulas studied use inside diameter of pipe, some in inches and some in feet. For ordinary design this inside diameter would depend upon the designer's judgment, but for Marine design the inside diameter is definite for various sized pipes for the various systems.

The Navy requires the following design pressures and they are suitable for all Marine design:

(*a*)	Fuel oil suction pipes, never subject to discharge pressure	50 p.s.i.
(*b*)	Fuel oil booster suction and transfer piping	150 p.s.i.
(*c*)	Fuel oil service pipes (service pump discharge)	350 p.s.i.
(*d*)	Fuel oil tank drain pump discharge	25 to 50 p.s.i.
(*e*)	Diesel fuel oil pump discharge	10 to 50 p.s.i.
(*f*)	Booster and transfer oil pump discharge	100 to 200 p.s.i.

The Navy also requires that all fuel oil piping shall be of steel and have dimensions in accordance with their standards. As all shipbuilders have standard pipe dimensions this resolves to the use of dimensions as given for 0-200 pounds working pressure for all low pressure services and of dimensions as given for 301-450 pounds working pressure for discharge from service pumps.

16. *Velocity of Flow (Item 3o):* This factor, in feet per second, appears in the basic formulas for both pressure drop and for Reynolds number but as mentioned above it may be converted into quantity if so desired.

However, to properly decide on size of pipe to be used for any particular design it is advisable to decide upon the allowable velocity. The Navy specifications formerly gave maximum allowable velocities of 6 feet per second for suction piping and 8 feet per second for discharge piping, but the present specifications do not give any limiting values and this factor is left to the judgment of the designer. For Marine design it will be found that to keep within allowable pressure drops, the velocities will be lower than the figures mentioned above.

After the specified quantity of oil has been reduced to G.P.M. the velocity can be obtained from the following:

$$\text{(XIII) Velocity (feet per second)} = \frac{\frac{\text{G.P.M.} \times 231}{60}}{\frac{3.1416 \times d^2}{4}} \times 1/12 =$$

$$\frac{\text{G.P.M.} \times 231 \times 4}{60 \times 3.1416 \times 12 \times d^2} = .4085 \times \frac{\text{G.P.M.}}{d^2}$$ in which d is the inside diameter of pipe in inches.

17. *Length of Pipe (Item 3p):* This factor, in feet, appears in all formulas for pressure drop, but does not affect the Reynolds number. In addition to the length of pipe involved, each fitting, valve, etc., in a pipe line adds to the friction drop and practically all of the articles studied agree that the most satisfactory method of including this additional friction loss is to add to the actual pipe length, an equivalent length for each fitting, etc., which would give the same friction drop as the fitting. The sum of the actual pipe length and of all of the equivalent lengths, in feet, is the factor "equivalent length" to be used in the formulas.

18. *Friction Loss Through Valves and Fittings (Item 3q):* Most of the references studied give the equivalent lengths for fittings, valves, etc., in terms of pipe diameters, but Crane Co.[11] give a chart from which the equivalent length for any fittings, etc., can be determined, for any size of pipe, in feet, but they note that this chart is based entirely on experimental data obtained from the flow of water and steam.

A tabulation of values of equivalent lengths as given by Nelson,[1] McAdams,[2] Danforth,[6-7] Crane,[11] Kent,[12] and Chemical Engineer-

ing [13] has been made and it was found that they differ to a great extent.

Another very important feature is the value of equivalent length to be used for turbulent and for stream line flow. Nelson [1] gives a table of "Commonly accepted equivalent lengths for turbulent conditions" and states that equivalent lengths for stream line flow are so small that they are usually neglected.

McAdams [2] gives a table of values of various fittings, etc., for turbulent flow and also curves for standard 90-degree elbows, based on Reynolds criterion. These curves show a fairly constant value through the turbulent flow region, a slight increase in the critical range and a decreasing value through the stream line region down to R = .002 and constant below that point. Danforth [6] gives a table of values and states: "This excess loss becomes less when handling very viscous material." Danforth [7] gives curves for various fittings, etc., showing a constant value through the turbulent region, an increasing value through the critical range and a uniformly decreasing value in the stream line region down to a minimum value. Crane [11] gives a very thorough study for water and steam but does not distinguish between turbulent and stream line flow. A later statement from the Crane Co. is to the effect that they believe values in the stream line range are lower and should be the same for all values of "R" below the critical. Chemical Engineering [13] gives information similar to McAdams.[2]

While the various references disagree as shown above, it seems logical that in the stream line range, the equivalent values would be lower than in the turbulent range, but that they would be higher for high values of "R" than for low values of "R" as the higher values of "R" correspond to higher values of velocity.

The various references studied did not stress the importance of this equivalent length as they were intended primarily for commercial design in which the actual length of pipe lines is very large compared to equivalent lengths for valves and fittings.

This equivalent length is the most uncertain of any of the variables in the formula for friction pressure drop and unfortunately for Marine design it is about the most important of these variables, as the pipe lines on board ship are comparatively short in comparison to the number of fittings and valves involved. In many

cases the equivalent lengths due to fittings, etc., are much greater than the actual length of pipe. With the difference in values given for the turbulent region and with the uncertainty of the values to be used for stream line flow, it would be possible to have errors of 100 per cent or more in the calculated friction drops.

Due to the importance of this factor for Marine design, the need for additional experimental data for fuel oil flow is evident.

19. *Friction Factor and Critical Range (Item 3r):* As stated above the friction factor (f) follows a definite law for stream line flow and most of the articles studied agree that it is a function of "R" (based on experimental data) for turbulent flow. Some of the articles have attempted to use a formula or a constant value for "f" in turbulent flow:

Arsenstein [13] gives for turbulent flow

(XIV) $f = \frac{.167}{\frac{(Vd)}{(v)}.17}$ in which v is kinematic viscosity in English units.

Kemler [14] gives tables of various values used by many investigators, which vary widely. He states that "any constant value of "f" would apply for a limited range of "R," but at low enough value of "R" it would seem to give a value less than that of smooth pipe."

As the great majority of articles studied give preference to a variable "f" for turbulent flow and as this preference seems to be justified, no further discussion will be attempted on the subject of a constant value or of one derived from a formula.

The basic experimental data was not investigated, as this has been done by many others and the results of their investigations are given in the articles studied. Most of these articles give the friction factor in the form of curves, based on values of "R" and these curves of course differ because of the units used. Kent [12] gives "f" in the form of a table, which requires interpolation. Some of the curves given are in the form of a single curve for all sizes of steel pipes, some give separate curves for different size pipes and some give many experimental curves. The following curves were finally selected for closer study—Nelson,[1] McAdams,[2] McAdams [4]

Danforth,[6] Crane,[11] and Kent [12] (table). These curves were analyzed with the following results, using Nelson [1] as a base.

Nelson [1] says that friction factors should be slightly higher than curve for small sizes (less than 1½ inch) and slightly less for pipes over 8 inches.

McAdams [2]—Friction drop formula resolves to same as Nelson's and uses same formula for "R". From the curves, the values of "f" for turbulent flow are the same as obtained from Nelson's curves, within the limits of reading these small scale curves. Values of "f" for stream line flow are calculated and agree with Nelson.

McAdams [4]—Friction drop formula resolves to same as Nelson's. A different formula is used for "R" but the curves are so drawn that the values of "f" for turbulent flow are the same as obtained from Nelson's curves, within the limitations of the small scale curves. Values of "f" for stream line flow are calculated and agree with Nelson.

Danforth [6]—Uses a friction drop formula which to give same results as Nelson requires "f" to be 54 times Nelson's values and a formula for "R" which resolves to 244.8 times Nelson's values. Separate curves are given for various size pipes and values were resolved to Nelson's base and the results indicated that in turbulent flow the values of "f" for 2-inch pipe agree fairly close with Nelson's values but are lower for larger pipes. Values of "f" for stream line are calculated and agree with Nelson.

Crane [11]—Uses a friction drop formula which to give same results as Nelson requires "f" to be four times Nelson's values and a formula for "R" which resolves to 7725 times Nelson's values. Separate curves are given for various size pipes and values were resolved to Nelson's base and the results indicated that in turbulent flow the value of "f" for small pipe agree fairly close with Nelson's values but are lower for larger pipes. Values of "f" for stream line flow are calculated and agree with Nelson.

Kent [12]—Similar to Crane [11] except that only one set of values are given and these are in the form of a table. Resolving the values to Nelson's base indicated that in turbulent flow the values of "f" are less than Nelson's values. Values of "f" for stream line flow are calculated and agree with Nelson. Kent [12] also gives the following conclusions:

(*a*) For brass pipe " f " varies only ± 5 per cent from an average value and is substantially independent of pipe size and fluid flowing for identical values of " R " in the range usually met in practice.

(*b*) For steel pipe " f " varies about ± 10 per cent from an average value and varies with size because roughness increases with pipe diameter.

(*c*) Effect of diameter is important as friction varies as $D.^5$

Kemler [14] states that " f " is independent of fluid for any value of " R ". His other conclusions are the same as Kent.

After resolving the various curves to the same base they were plotted on the same sheet and from this sheet and the above notes a final design curve may be determined upon.

As stated above there is a critical range above the point of critical velocity which should be avoided in design. This critical range is variously defined and located in terms of " R ".

Nelson [1]—Does not define but from curves of " f " the critical range is located between values of " R " of about .123 to .16 (c.g.s. system).

McAdams [2]—Discusses possibility of stream line flow above the critical velocity and turbulent flow below. He advises the use of

$$\text{critical velocity} = \text{Uc} = \frac{.122 \times z}{\text{Ds}} \text{ (c.g.s.)}$$

and when velocity is above this point the use of turbulent flow values of " f " and when below the use of stream line values. This critical velocity gives about the same result as shown on Nelson's curve.

McAdams [4]—Gives critical range between values of " R " of 2100 to 2300 (English units), equivalent to c.g.s. of .27 to .29.

Kinney [5] uses turbulent values above " R " of 65, equivalent to c.g.s. of .26.

Danforth [6] uses turbulent values above " R " of 57, equivalent to c.g.s. of .23.

Crane [11] does not definitely locate the critical range but uses stream line value of " f " below " R " of 1000, equivalent to c.g.s. of .129.

Kent [12] gives critical range between values of " R " of 2000 to 3000, equivalent to c.g.s. of .26 to .4.

Chemical Engineering [13] gives critical range between values of "R" of 2100 to 2300, equivalent to c.g.s. of .27 to .29.

Kemler [14] uses turbulent values above "R" of 2000, equivalent to c.g.s. of .26.

Pigott [16] gives unstable region between values of "R" of 1200 to 2500, equivalent to c.g.s. of .15 to .32.

20. *Type, Size and Roughness of Pipe (Item 3s):* The effect of size and roughness of pipe on the friction factor is the subject of much comment. These comments vary and agree only on the fact that roughness does tend to increase friction factor and that this effect is much greater in small pipes than in larger ones. Some of the articles give curves and tables in an attempt to make this definite but most of them agree that as there is no method of evaluating roughness it must be left to the judgment of the designer and furthermore most of them state that friction factor curves are for commercially clean pipe and are sufficiently accurate for practical use. Even where curves are given they do not show any serious effect for the probable sizes of clean steel pipes within the probable range of "R" for Marine work.

It is, therefore, felt that for purposes of standard data, the effect of roughness may be neglected especially as clean pipe is used and as the oils used will not cause much tuberculation. Also for Marine design most calculations will fall within stream line flow or in low values of "R" for turbulent flow and as the curves for all sizes of pipe practically meet at these low values, it is felt that an average curve of "f" may be used for all sizes.

21. *Friction Pressure Drop:* There are no definite specifications for pressure drop limits. However, it is good practice to have a discharge pressure of 350 pounds per square inch at the service pumps and a pressure at inlet to boiler front manifolds of not less than 300 pounds per square inch, which gives the allowable maximum pressure drop for service pump discharge as 50 pounds per square inch, including drop through fuel oil heater and including static head. The pressure drop through the heater should not exceed 30 pounds.

Owing to the fact that pumps are not 100 per cent efficient the maximum allowable pressure drop for suction pipes should not exceed about 11 pounds per square inch (22.46 inches mercury) including static lift.

The allowable pressure drop for other discharge piping must be left to the judgment of the designer.

All of the articles studied give formulas for calculating the friction pressure drop and some of them also give curves designed to eliminate the necessity of calculations. These two methods will be discussed separately.

22. *Friction Pressure Drop Formula:* It is possible to reduce all of the various formulas given to the same basic formula regardless of nomenclature or units used. The following tabulation is made on the basis of reducing all the formulas to the form used by Nelson,[1] which agrees with the basic formulas (I), (III), and (IV). For purposes of comparison, all of the formula will be given in the following nomenclature regardless of the nomenclature used in the various articles.

NOMENCLATURE.

P = Pressure Drop in Pounds per square inch

PF = Pressure Drop in Pounds per square foot = $P \times 144$

PH = Pressure Drop in Feet of Head = $\frac{P \times 144}{62.3 \times s}$.

L = Equivalent Length in Feet = Actual Pipe Length + equivalent lengths for fittings, valves, etc.

D = Inside diameter of pipe in feet = d/12

d = Inside diameter of pipe in inches = $D \times 12$

Gm = Gallons per minute

Gh = Gallons per hour

B = Barrels per hour

V = Velocity in Feet per second

s = Specific Gravity

p = Density (Pounds per cubic feet) = $62.3 \times s$

z = Absolute viscosity in centipoises = $U \times 100$

U = Absolute viscosity in poises = $\frac{z}{100}$.

μ = Absolute viscosity in poundals = $.000672 \times z$

y = Kinematic Viscosity $= \frac{\mu}{p}$ or $\frac{z}{s}$.

s.v. = Specific Volume $= 1/p$
R = Reynolds Number or Reynolds Criterion
f = Friction Factor

FORMULA.

Nelson [1]

(XV) Turbulent Flow: $P = \frac{.323 \times f \times s \times V^2 \times L}{d}$, using $R = \frac{dVs}{z}$ and obtaining " f " from curves of " R."

Stream Line: Using $f = \frac{.00207 \times z}{dVs} = \frac{.00207}{R}$ $P = \frac{.000669VLz}{d^2}$.

McAdams [2]

Turbulent Flow: $PF = \frac{4 \times f \times p \times L \times V^2}{2gD}$, or converting $P = \frac{4 \times f \times p \times L \times V^2}{2gD \times 144}$, using $R = \frac{dVs}{z}$ and obtaining " f " from curves of " R."

This resolves to: $P = \frac{.323 \times f \times s \times V^2 \times L}{d}$.

Stream Line: $PF = \frac{32\mu LV}{gD^2}$, which resolves to $P = \frac{.000668VLz}{d^2}$, or using $f = \frac{.00207 \times z}{dVs}$ in turbulent formula will give same result.

These resolved formulas are same as Nelson.

McAdams [4]

Turbulent Flow: $PF = \frac{4 \times f \times p \times V^2 \times L}{2gD}$, or converting $P = \frac{4 \times f \times p \times V^2 \times L}{2gD \times 144}$, using $R = \frac{DVp}{\mu}$ and obtaining "f" from curves of "R."

This resolves to: $P = \frac{.323 \times f \times s \times V^2 \times L}{d}$.

Stream Line: Using $f = 16/R$, $P = \frac{.000669VLz}{d^2}$.

These resolved formulas are same as Nelson.

Kent [12]

Turbulent Flow: $PH = \frac{fLV^2}{2gD}$, or converting $P = \frac{f \times p \times L \times V^2}{2gD \times 144}$, using $R = \frac{DVp}{\mu}$ and obtaining "f" from table of "R."

This resolves to: $P = \frac{.0806 \times f \times s \times V^2 \times L}{d}$, which is same as Nelson except that "f" must be $\frac{.323}{.0806} = 4$ times the value of "f" used by Nelson.

Stream Line: Using $f = 64/R$, $P = \frac{2 \times \mu \times L \times V}{9gD^2}$ $= \frac{.000669VLz}{d^2}$, which is the same as Nelson.

Crane [11]

Turbulent Flow: $P = \frac{f \times L \times V^2}{772 \times d \times sv} = \frac{f \times p \times L \times V^2}{772d}$,

using $R = \frac{DVp}{\mu}$ and obtaining " f " from curves of " R."

This resolves to: $P = \frac{.0807 \times f \times s \times V^2 \times L}{d}$, which is same as Nelson except that " f " must be $\frac{.323}{.0806} = 4$ times the value of " f " used by Nelson.

Stream Line: $PF = \frac{143.3 \times \mu \times L \times V}{d^2}$, which resolves to: $P = \frac{143.3 \times \mu \times L \times V}{144 d^2}$ or $P = \frac{2\mu LV}{9gD^2} = \frac{.000668VLz}{d^2}$, which is the same as Nelson. Using $f = 64/R$ in turbulent formula will give same result.

Kinney [5]

Turbulent Flow: $P = \frac{fsGm^2}{d^5}$ per 1000 feet $= \frac{f \times s \times Gm^2 \times L}{d^5 \times 1000}$, using $R = \frac{SGm}{dU}$ and obtaining " f " from curves of " R."

As $Gm = \frac{Vd^2}{.4085}$, this resolves to: $P = \frac{.006 \times f \times s \times V^2 \times L}{d}$, which is same as Nelson except that " f " must be $\frac{.323}{.006} = 54$ times the value of " f " used by Nelson.

Stream Line: $P = \frac{27.3 \times U \times Gm}{d^4}$ per 1000 feet $= \frac{27.3 \times U \times Gm \times L}{d^4 \times 1000}$ for values of R less than 65.

This resolves to: $P = \frac{.000668VLz}{d^2}$, which is same as Nelson.

Danforth [6]

Same as Kinney except that stream line flow only applies for values of R less than 57.

Beck [17]

Turbulent Flow: $P = \frac{.558 \times f \times B^2 \times p}{d^5}$ per mile $= \frac{.558 \times f \times p \times B^2 \times L}{d^5 \times 5280}$, using $R = \frac{VDp}{\mu}$ and obtaining "f" from curves of "R."

This resolves to: $P = \frac{.0806 \times f \times s \times V^2 \times L}{d}$, which is same as Kent.

Lubrication [8]

Same as Nelson.

Kemler [14]

Turbulent Flow: $PF = \frac{f \times L \times V^2 \times p}{2gD}$, or converting $P = \frac{f \times L \times V^2 \times p}{2gD \times 144}$, using $R = \frac{DVp}{\mu}$ and obtaining "f" from various curves.

This resolves to: $P = \frac{.0806 \times f \times s \times V^2 \times L}{d}$, which is the same as Kent.

Stream Line: $PF = \frac{64 \times \mu \times L \times V}{2gD^2}$ or $P = \frac{64 \times \mu \times L \times V}{2gD^2 \times 144}$, which resolves to

$P = \frac{.000668 \times V \times L \times z}{d^2}$, which is the same as Nelson.

Using $f = 64/R$ in turbulent formula will give same result.

Pigott [16]

Turbulent Flow: $P = \frac{.000108 \times f \times L \times p \times V^2}{D}$ or $\frac{.0067 \times f \times s \times V^2 \times L}{D}$, using $R = \frac{DVp}{\mu}$ and obtaining "f" from various curves.

This resolves to: $P = \frac{.0808 \times f \times s \times V^2 \times L}{d}$, which is same as Kent.

Stream Line: $P = \frac{.00691 \times \mu \times L \times V}{D^2}$, which resolves to $P = \frac{.000669 \times V \times L \times z}{d^2}$, which is the same as Nelson.

Aisenstein [15]

Gives a formula for turbulent flow which eliminates the friction factor: $PH = \frac{f \times L \times V^2}{2gD}$ or $P = \frac{f \times p \times L \times V^2}{2gD \times 144}$

using $f = \frac{.167}{\frac{(VD)}{(y)}^{.17}}$ (Par. 19) gives $P = .329\ y^{.17} \times p \times \frac{V^{1.183}}{d^{1.17}}$ per 1000 feet.

Neglecting the last formula (Aisenstein), it is seen that all of these formulas will give the same result for stream line flow and the same results for turbulent flow if the values of "f" are equivalent. (See Paragraph 19.)

It will be further seen that the selection of a formula for standard use depends entirely upon the units to be used and the friction factor curve selected. This will be further discussed under "Conclusions."

23. *Friction Pressure Drop Curves:* Several of the articles give curves from which pressure drops may be obtained without the necessity of making calculations.

Crane [11] gives a curve sheet from which can be taken the pressure drop in pounds per square inch or in feet if there is known the quantity of oil (G.P.M., BBLs per Hour or BBLs per min.) inside diameter of pipe in inches, Saybolt Universal viscosity or kinematic viscosity in $\frac{cm^2}{SEC}$, degrees A.P.I. or Specific gravity and the length in feet.

Danforth [6] gives separate curves for various sizes of standard pipes, from which can be taken the pressure in pounds per square inch if there is known the quantity of oil (G.P.M. or BBLs per Hour), Saybolt Universal viscosity or kinematic viscosity $= \frac{\text{poises}}{\text{sp. gr.}}$, Specific gravity or degrees Baume and length.

Beck [17] gives nomographic charts for determining "R" and friction loss.

Chemical Engineer [13] gives alignment charts for friction loss.

These curves seem to give very good results but they are necessarily on a small scale and attempt to cover all cases. As will be shown later under "Conclusions" it is possible to make much simpler curves to cover the standard conditions involved in Marine design.

CONCLUSIONS FOR MARINE DESIGN.

1. *General Design.*

(*a*) Basic Oils—Navy grade fuel oil—65 S.S.F. at 122 degrees F.—8 degrees A.P.I. Bunker "C" fuel oil—300 S.S.F. at 122 degrees F.—2 degrees A.P.I. Diesel fuel oil—45 S.S.U. at 100 degrees F.—25 degrees A.P.I.

(*b*) Pumping Conditions—F.O. Booster and Transfer piping and Service Pump suctions and discharge to fuel oil heaters—700 S.S.F. (Equivalent to a pumping temperature of about 65 degrees F. for Navy grade and about 105 degrees F. for Bunker "C").

Discharge from fuel oil heaters to boiler front manifolds—20 S.S.F. (Equivalent to a pumping temperature of about 180 degrees F. for Navy grade and about 225 degrees F. for Bunker "C").

Fuel Oil Tank Drain Pump piping same as F.O. Booster pump piping.

Diesel fuel oil piping—70 S.S.U. (Equivalent to a pumping temperature of about 50 degrees F.).

(*c*) Design Pressures—F.O. Booster and Transfer Pump Piping and Service Pump suctions—50 to 200 pounds per square inch gage. Service Pump discharge—350 pounds per square inch gage. F.O. Tank Drain Pump Discharge—25-50 pounds per square inch gage. Diesel F.O. piping—10-50 pounds per square inch gage.

(*d*) F.O. Pipe material—Steel.

(*e*) F.O. Pipe dimension—As given in Navy Department standards or special shipbuilders' standards. Use 0-200 pounds W.P. for all low pressure systems and 301-450 pounds W.P. for all high pressure piping.

(*f*) Pressure Drop Limits—Suction piping 11 pounds per square inch.

Discharge from Service Pumps—50 pounds per square inch, including drop through heater.

All other discharge piping left to judgment of designer.

Detail Design General: Before outlining design data for standard Marine conditions, the following design method is proposed and is suitable for any fuel oil friction problem either standard or special and may also be used for lubricating oil piping.

To properly decide on size of pipe to be used for any particular design, it is necessary to consider the velocity of flow and the Navy Department specifications require that pipe line velocities be shown on diagrammatic plans of fuel oil systems. It thus becomes necessary to calculate the velocity and therefore the standard formula proposed is based upon velocity, thus eliminating the factor of

diameter to the fifth power. Also the proposed formula will give friction pressure drop in terms of pounds per square inch, which can readily be converted into inches of mercury for suction lines. As most conversion charts use centipoises and to be in line with other Navy Department design data, the c.g.s. units will be used for viscosities.

NOMENCLATURE.

P = Friction Pressure Drop in Pounds per square inch
d = Diameter of Pipe in inches (inside)
L = Equivalent Length of Pipe in feet
S = Specific Gravity of Basic Oil
s = Specific Gravity of Oil at Pumping Conditions
z = Absolute Viscosity in Centipoises at Pumping Conditions

(XVI)

$$y = \text{Kinematic Viscosity at Pumping Conditions} = \frac{z}{s}$$

W = Pounds of Oil per hour
G = Gallons per minute at Pumping Conditions
V = Velocity of Flow in Feet per Second

(XVII)

$$R = \text{Reynolds Criterion} = \frac{dVs}{z} = \frac{dV}{y}$$

f = Friction Factor

FORMULA.

Nelson's [1] form of formula (XV) will be used:

$$\text{(XV)}\quad P = \frac{.323 \times f \times s \times V^2 \times L}{d}$$

The various factors in this friction pressure drop formula may be obtained and are to be used as shown in following discussion.

(*a*) *Friction Factor* (*f*): This factor is a function of the Reynolds criterion ($R = \frac{dV}{y}$). In stream line flow it follows a definite mathematical law and may be taken as

$$\text{(XVIII)}\quad f = \frac{.00207 \times z}{dVs} = \frac{.00207 \times y}{dV} = \frac{.00207}{R}$$

In turbulent flow this factor is so much affected by various outside conditions, that it is not readily reducible to a simple formula, but is based on experimental data and is given in the form of curves having values of "R" as abscissa and values of "f" as ordinates.

There are available the results of many experiments giving different values of "f," but this data has been tabulated and put into the form of average curves by most of the articles studied. Figure 1 shows a comparison of a number of these average curves. As most of the calculations for Marine design will fall within stream line flow or in low values of "R" for turbulent flow it is proposed to use one curve for all sizes of pipes and in order to avoid consideration of roughness it is further proposed to use the higher curve as given by Nelson [1] and which gives same results as McAdams [2-4] and this curve is shown on Figure 2. For convenience Figure 2 also includes the calculated values of "f" in stream line flow.

The critical velocity, below which stream line conditions should be used, is indicated on this curve at R = .123 and it is considered that the critical range will extend up to about R = .2 and this critical range should be avoided in design.

(*b*) *Viscosity:* The only viscosimeter scales to be considered here are the Saybolt Universal and Furol. These are arbitrary scales and cannot be used directly in formula XVII for Reynolds criterion. The actual unit of viscosity is absolute viscosity (z) which will be used in centipoises. The most convenient unit for use, as shown by formula (XVII) is kinematic viscosity (y) which is the ratio between absolute viscosity and specific gravity ($y = \frac{z}{s}$).

This kinematic viscosity has a definite relation to viscosimeter scales and may be calculated from formula (VII) and (VIII).

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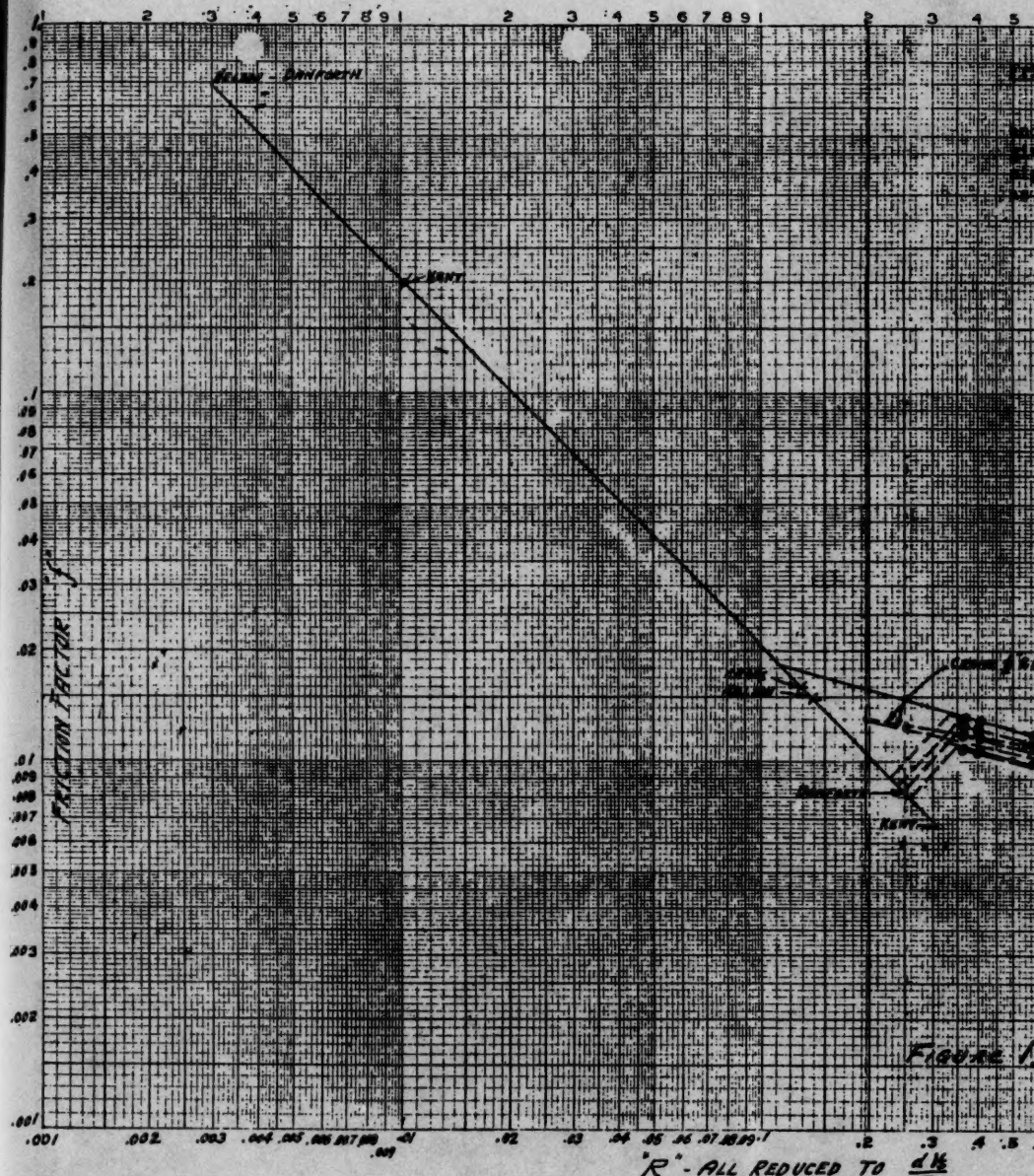
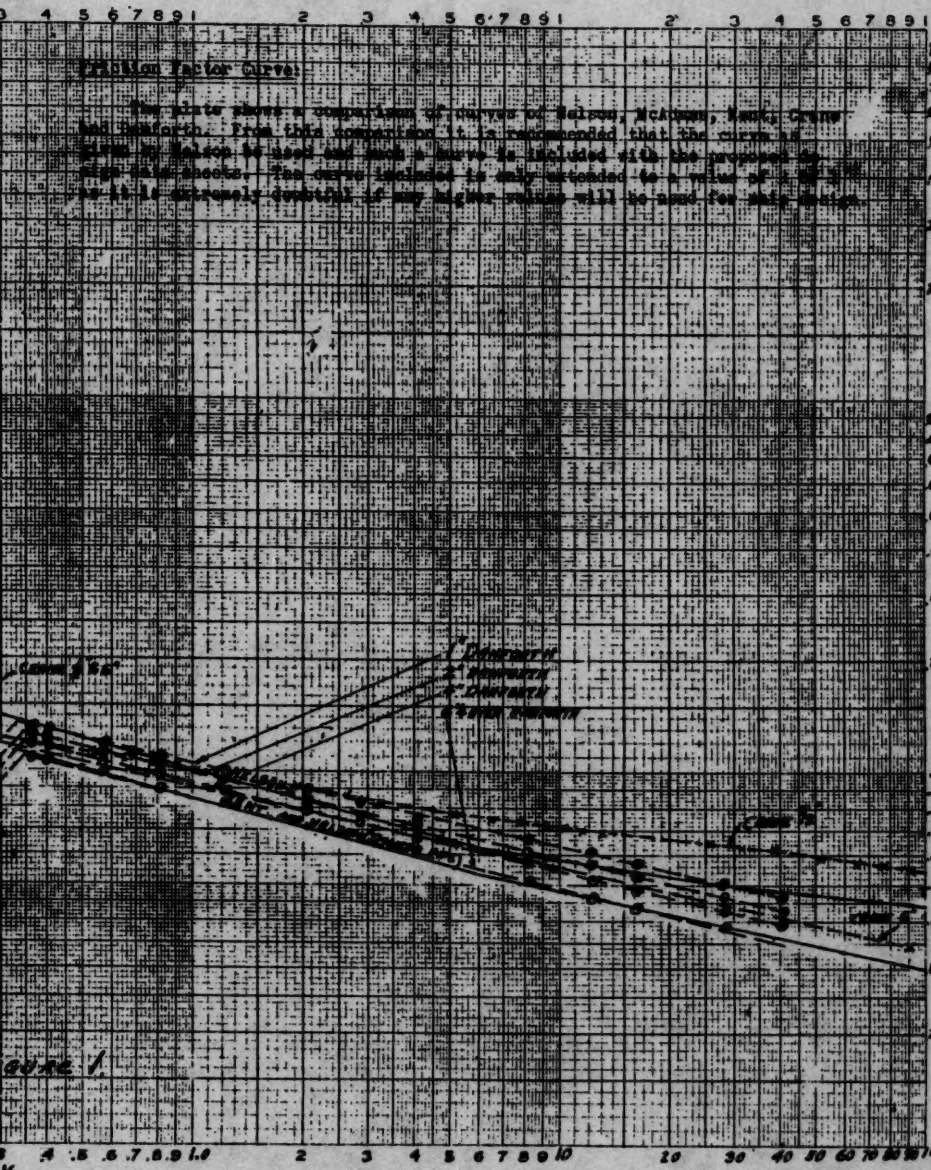


FIGURE


Friction Factor Curve.
The plate shows a comparison of curves by Nelson, McAdams, Mantz, Crane and Kenforth. From this comparison it is recommended that the curves by Nelson be used and that a curve be included with the Reynolds number to .1 decimals. The curve included is only extended to a value of 2 x 10^6 for it is extremely doubtful if any higher values will be used for duct design.
FIGURE 1.

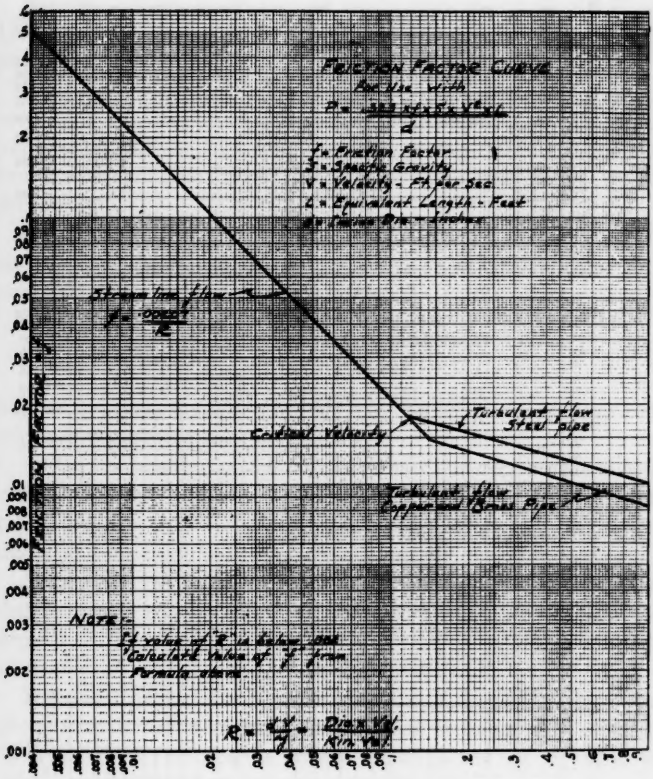


FIGURE 2.

(VII) Saybolt Universal—$y = .220 \times \text{S.S.U.} - \frac{180}{\text{SSU}}$

(VIII) Saybolt Furol—$y = 2.04 \times \text{S.S.F.} - \frac{160}{\text{SSF}}$

In order to convert viscosity changes to temperature changes, use Figure 3, which is taken from Viscosity-Temperature Diagram [3] (Form N. Eng. 264). This diagram can also be used to convert S.S.U. to S.S.F. and to obtain values of "y."

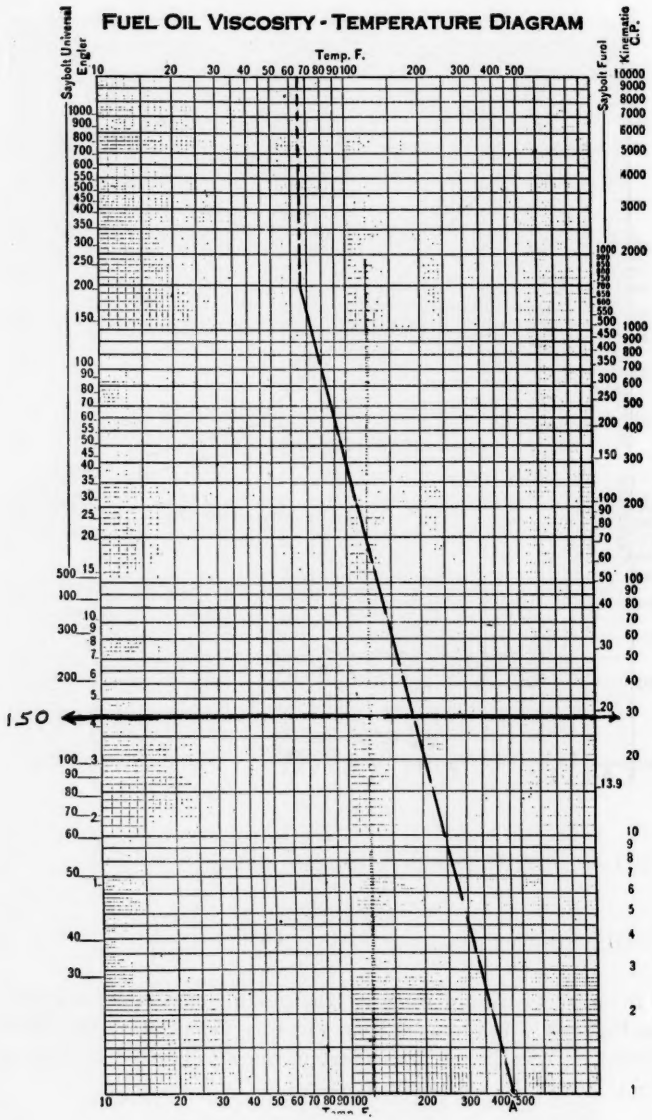

FUEL OIL VISCOSITY - TEMPERATURE DIAGRAM
Temp. F.
Saybolt Universal Engler
Saybolt Furol
Kinematic C.P.
150
A

FIGURE 3.

Example: Navy grade fuel oil has a viscosity of 65 S.S.F. at 122 degrees F. and is to be pumped at 700 S.S.F. What is the pumping temperature?

Solution: From point "A" (Figure 3) draw a line to the intersection of 65 S.S.F. and 122 degrees F. Follow this line to its intersection with the horizontal line representing 700 S.S.F. and the ordinate at this point gives the corresponding temperature of 65 degrees F.

(*c*) *Specific Gravity* (*s*): The relation between degrees A.P.I. and specific gravity (S) of basic oil may be calculated from formula IX.

(IX) Specific Gravity (S) $= \frac{141.5}{^{\circ}\text{A.P.I.} + 131.5}$ or may readily be obtained from the table given in A.S.T.M.[10] standards, page 156, which is reproduced here as Figure 4.

As the temperature of the oil is increased, or decreased, the specific gravity (s) at pumping conditions may be calculated from tables given in A.S.T.M.[10] standards, page 314, which are reproduced here as Figures 5, 5A, 5B, 5C and 5D.

Example: Navy grade fuel oil has a basic A.P.I. gravity of 8 degrees and a corresponding specific gravity (S) of 1.0143 at 60 degrees F. (Figure 4). It is to be pumped at a temperature of 65 degrees F. What is the final specific gravity (s)?

Solution: From table 5 it is seen that 8 degrees A.P.I. is in group O and from this group the multiplier is found to be .9983 at 65 degrees F. Then specific gravity (s) = S $\times$.9983 = 1.0143 $\times$ .9983 = 1.0126.

(*d*) *Quantity of Oil:* If the specified quantity of oil is given as pounds per hour, it can readily be converted to gallons per minute by the use of Figures 4 and 5.

Example: If "W" is the pounds per hour of Navy grade fuel oil to be pumped at a temperature of 180 degrees F., what will be the G.P.M. (G) at pumping conditions?

Solution: Basic oil has an A.P.I. gravity of 8 degrees corresponding to a specific gravity (S) of 1.0143. From Table 4 this is equivalent to .1184 gallons per pound at 60 degrees F. From Table 5, the multiplier is found to be .9594 at 180 degrees F. Then G =

The following table is primarily an abridgement of Table 5 of the "National Standard Petroleum Oil Tables" as published in the U. S. Bureau of Standards *Circular No. 154*, which contains similar data for each 0.1° A.P.I. from 10 to 100° A.P.I., inclusive. The data for gravities from 0 to 9° A.P.I., inclusive, have been added in order to extend the use of the A.P.I. gravity scale to petroleum oils having a specific gravity slightly greater than unity.

TABLE SHOWING SPECIFIC GRAVITIES, POUNDS PER GALLON, AND GALLONS PER POUND AT 60° F., CORRESPONDING TO DEGREES A.P.I.

Degrees A. P. I.	Specific Gravity at 60°/60° F.	Pounds per Gallon at 60° F.[1]	Gallons per Pound at 60° F.[1]	Degrees A. P. I.	Specific Gravity at 60°/60° F.	Pounds per Gallon at 60° F.[1]	Gallons per Pound at 60° F.[1]
0	1.0760	8.962	0.1116	50	0.7796	6.490	0.1541
1	1.0679	8.895	0.1124	51	0.7753	6.455	0.1549
2	1.0599	8.828	0.1133	52	0.7711	6.420	0.1558
3	1.0520	8.762	0.1141	53	0.7669	6.385	0.1566
4	1.0443	8.698	0.1150	54	0.7628	6.350	0.1575
5	1.0366	8.634	0.1158	55	0.7587	6.316	0.1583
6	1.0291	8.571	0.1167	56	0.7547	6.283	0.1592
7	1.0217	8.509	0.1175	57	0.7507	6.249	0.1600
8	1.0143	8.448	0.1184	58	0.7467	6.216	0.1609
9	1.0071	8.388	0.1192	59	0.7428	6.184	0.1617
10	1.0000	8.328	0.1201	60	0.7389	6.151	0.1626
11	0.9930	8.270	0.1209	61	0.7351	6.119	0.1634
12	0.9861	8.212	0.1218	62	0.7313	6.087	0.1643
13	0.9792	8.155	0.1226	63	0.7275	6.056	0.1651
14	0.9725	8.099	0.1235	64	0.7238	6.025	0.1660
15	0.9659	8.044	0.1243	65	0.7201	5.994	0.1668
16	0.9593	7.989	0.1252	66	0.7165	5.964	0.1677
17	0.9529	7.935	0.1260	67	0.7128	5.934	0.1685
18	0.9465	7.882	0.1269	68	0.7093	5.904	0.1694
19	0.9402	7.830	0.1277	69	0.7057	5.874	0.1702
20	0.9340	7.778	0.1286	70	0.7022	5.845	0.1711
21	0.9279	7.727	0.1294	71	0.6988	5.817	0.1719
22	0.9218	7.676	0.1303	72	0.6953	5.788	0.1728
23	0.9159	7.627	0.1311	73	0.6919	5.759	0.1736
24	0.9100	7.578	0.1320	74	0.6886	5.731	0.1745
25	0.9042	7.529	0.1328	75	0.6852	5.703	0.1753
26	0.8984	7.481	0.1337	76	0.6819	5.676	0.1762
27	0.8927	7.434	0.1345	77	0.6787	5.649	0.1770
28	0.8871	7.387	0.1354	78	0.6754	5.622	0.1779
29	0.8816	7.341	0.1362	79	0.6722	5.595	0.1787
30	0.8762	7.296	0.1371	80	0.6690	5.568	0.1796
31	0.8708	7.251	0.1379	81	0.6659	5.542	0.1804
32	0.8654	7.206	0.1388	82	0.6628	5.516	0.1813
33	0.8602	7.163	0.1396	83	0.6597	5.491	0.1821
34	0.8550	7.119	0.1405	84	0.6566	5.465	0.1830
35	0.8498	7.076	0.1413	85	0.6536	5.440	0.1838
36	0.8448	7.034	0.1422	86	0.6506	5.415	0.1847
37	0.8398	6.993	0.1430	87	0.6476	5.390	0.1855
38	0.8348	6.951	0.1439	88	0.6446	5.365	0.1864
39	0.8299	6.910	0.1447	89	0.6417	5.341	0.1872
40	0.8251	6.870	0.1456	90	0.6388	5.316	0.1881
41	0.8203	6.830	0.1464	91	0.6360	5.293	0.1889
42	0.8155	6.790	0.1473	92	0.6331	5.269	0.1898
43	0.8109	6.752	0.1481	93	0.6303	5.246	0.1906
44	0.8063	6.713	0.1490	94	0.6275	5.222	0.1915
45	0.8017	6.675	0.1498	95	0.6247	5.199	0.1924
46	0.7972	6.637	0.1507	96	0.6220	5.176	0.1932
47	0.7927	6.600	0.1515	97	0.6193	5.154	0.1940
48	0.7883	6.563	0.1524	98	0.6166	5.131	0.1949
49	0.7839	6.526	0.1532	99	0.6139	5.109	0.1957
				100	0.6112	5.086	0.1966

[1] The data in this column are based upon apparent weights in air, the weight of one gallon of water at 60° F. in air being assumed to be 8.32828 lb. Consequently, a correction for the buoyancy of air is required in computing these figures from the specific gravities in the second column

FIGURE 4.

STANDARD ABRIDGED VOLUME CORRECTION TABLE FOR PETROLEUM OILS[1]

This table is issued under the fixed designation D 206; the final number indicates the year of original adoption as standard or, in case of revision, the year of last revision.

This abridged table has been prepared by the U. S. Bureau of Standards to meet a demand from the oil industry for a short and convenient table for reducing oil volumes to the basis of 60 F., when extreme accuracy is not required. It is not intended to replace the Bureau of Standards *Circular No. 154*, but rather to supplement it and especially to replace the various abridged tables heretofore employed in the oil industry.

This abridged table includes all of the figures appearing in the table as issued in 1925, and, in addition, two new groups. The figures for Group 0 are based on Table I of the Bureau of Standards *Miscellaneous Publication No. 97*, and the figures for Group 7 are based on the Bureau of Standards *Circular No. 154*. This extension of the original abridged table has been rendered necessary on account of the growing importance of very heavy and very light petroleum products.

The groups, coefficients of expansion, gravity (degrees A.P.I.), and gravity ranges for the several subdivisions of the present abridged table follow:

Group Number	Coefficient of Expansion at 60 F.	Gravity, Degrees A.P.I.	Gravity Range of Group (Degrees A.P.I. at 60 F.)
0	0.00035	..	Up to 14.9
1	0.00040	22	15.0 to 34.9
2	0.00050	44	35.0 to 50.9
3	0.00060	58	51.0 to 63.9
4	0.00070	72	64.0 to 78.9
5	0.00080	86	79.0 to 88.9
6	0.00085	91	89.0 to 93.9
7	0.00090	97	94.0 to 100.0

Figure 5.

Standard Abridged Volume Correction Table for Petroleum Oils

GROUP 0 [a]

Legend: t = observed temperature in degrees Fahrenheit; M = multiplier for reducing oil volumes to the basis of 60 F.

t	M	t	M	t	M	t	M	t	M
0	1.0205	50	1.0034	100	0.9864	150	0.9695	200	0.9527
1	1.0202	51	1.0031	101	0.9860	151	0.9691	201	0.9523
2	1.0198	52	1.0027	102	0.9857	152	0.9688	202	0.9520
3	1.0195	53	1.0024	103	0.9854	153	0.9684	203	0.9516
4	1.0191	54	1.0020	104	0.9850	154	0.9681	204	0.9513
5	1.0188	55	1.0017	105	0.9847	155	0.9678	205	0.9510
6	1.0185	56	1.0014	106	0.9844	156	0.9674	206	0.9506
7	1.0181	57	1.0010	107	0.9840	157	0.9671	207	0.9503
8	1.0178	58	1.0007	108	0.9837	158	0.9668	208	0.9500
9	1.0174	59	1.0003	109	0.9833	159	0.9664	209	0.9496
10	1.0171	60	1.0000	110	0.9830	160	0.9661	210	0.9493
11	1.0167	61	0.9997	111	0.9827	161	0.9658	211	0.9490
12	1.0164	62	0.9993	112	0.9823	162	0.9654	212	0.9486
13	1.0161	63	0.9990	113	0.9820	163	0.9651	213	0.9483
14	1.0157	64	0.9986	114	0.9816	164	0.9647	214	0.9480
15	1.0154	65	0.9983	115	0.9813	165	0.9644	215	0.9476
16	1.0150	66	0.9980	116	0.9810	166	0.9641	216	0.9473
17	1.0147	67	0.9976	117	0.9806	167	0.9637	217	0.9470
18	1.0143	68	0.9973	118	0.9803	168	0.9634	218	0.9466
19	1.0140	69	0.9969	119	0.9799	169	0.9631	219	0.9463
20	1.0137	70	0.9966	120	0.9796	170	0.9627	220	0.9460
21	1.0133	71	0.9963	121	0.9793	171	0.9624	221	0.9456
22	1.0130	72	0.9959	122	0.9789	172	0.9621	222	0.9453
23	1.0126	73	0.9956	123	0.9786	173	0.9617	223	0.9450
24	1.0123	74	0.9952	124	0.9783	174	0.9614	224	0.9446
25	1.0120	75	0.9949	125	0.9779	175	0.9610	225	0.9443
26	1.0116	76	0.9945	126	0.9776	176	0.9607	226	0.9440
27	1.0113	77	0.9942	127	0.9772	177	0.9604	227	0.9436
28	1.0109	78	0.9939	128	0.9769	178	0.9600	228	0.9433
29	1.0106	79	0.9935	129	0.9766	179	0.9597	229	0.9430
30	1.0102	80	0.9932	130	0.9762	180	0.9594	230	0.9426
31	1.0099	81	0.9928	131	0.9759	181	0.9590	231	0.9423
32	1.0095	82	0.9925	132	0.9755	182	0.9587	232	0.9420
33	1.0092	83	0.9922	133	0.9752	183	0.9584	233	0.9416
34	1.0089	84	0.9918	134	0.9749	184	0.9580	234	0.9413
35	1.0085	85	0.9915	135	0.9745	185	0.9577	235	0.9410
36	1.0082	86	0.9912	136	0.9742	186	0.9573	236	0.9406
37	1.0079	87	0.9908	137	0.9739	187	0.9570	237	0.9403
38	1.0075	88	0.9905	138	0.9735	188	0.9567	238	0.9400
39	1.0072	89	0.9901	139	0.9732	189	0.9563	239	0.9396
40	1.0068	90	0.9898	140	0.9728	190	0.9560	240	0.9393
41	1.0065	91	0.9894	141	0.9725	191	0.9557	241	0.9390
42	1.0061	92	0.9891	142	0.9722	192	0.9553	242	0.9386
43	1.0058	93	0.9888	143	0.9718	193	0.9550	243	0.9383
44	1.0054	94	0.9884	144	0.9715	194	0.9547	244	0.9380
45	1.0051	95	0.9881	145	0.9711	195	0.9543	245	0.9376
46	1.0048	96	0.9878	146	0.9708	196	0.9540	246	0.9373
47	1.0044	97	0.9874	147	0.9705	197	0.9537	247	0.9370
48	1.0041	98	0.9871	148	0.9701	198	0.9533	248	0.9367
49	1.0038	99	0.9867	149	0.9698	199	0.9530	249	0.9363

Figure 5A.

STANDARD ABRIDGED VOLUME CORRECTION TABLE FOR PETROLEUM OILS

GROUP 0 ° (*Continued*)

Legend: t = observed temperature in degrees Fahrenheit; M = multiplier for reducing oil volumes to the basis of 60 F

t	M	t	M	t	M	t	M	t	M
250	0.9360	300	0.9195	350	0.9031	400	0.8869	450	0.8709
251	0.9356	301	0.9191	351	0.9028	401	0.8865	451	0.8706
252	0.9353	302	0.9189	352	0.9025	402	0.8863	452	0.8703
253	0.9350	303	0.9185	353	0.9021	403	0.8860	453	0.8700
254	0.9347	304	0.9181	354	0.9018	404	0.8856	454	0.8697
255	0.9343	305	0.9178	355	0.9015	405	0.8853	455	0.8693
256	0.9340	306	0.9175	356	0.9012	406	0.8850	456	0.8690
257	0.9337	307	0.9172	357	0.9008	407	0.8847	457	0.8687
258	0.9333	308	0.9168	358	0.9005	408	0.8843	458	0.8684
259	0.9330	309	0.9165	359	0.9002	409	0.8840	459	0.8681
260	0.9327	310	0.9162	360	0.8999	410	0.8837	460	0.8678
261	0.9323	311	0.9158	361	0.8995	411	0.8834	461	0.8674
262	0.9320	312	0.9155	362	0.8992	412	0.8831	462	0.8671
263	0.9317	313	0.9152	363	0.8989	413	0.8827	463	0.8668
264	0.9313	314	0.9149	364	0.8986	414	0.8824	464	0.8665
265	0.9310	315	0.9145	365	0.8982	415	0.8821	465	0.8662
266	0.9307	316	0.9142	366	0.8979	416	0.8818	466	0.8659
267	0.9303	317	0.9139	367	0.8976	417	0.8815	467	0.8655
268	0.9300	318	0.9135	368	0.8973	418	0.8811	468	0.8652
269	0.9297	319	0.9132	369	0.8969	419	0.8808	469	0.8649
270	0.9294	320	0.9129	370	0.8966	420	0.8805	470	0.8646
271	0.9290	321	0.9126	371	0.8963	421	0.8802	471	0.8643
272	0.9287	322	0.9122	372	0.8960	422	0.8799	472	0.8640
273	0.9284	323	0.9119	373	0.8956	423	0.8795	473	0.8636
274	0.9280	324	0.9116	374	0.8953	424	0.8792	474	0.8633
275	0.9277	325	0.9113	375	0.8950	425	0.8789	475	0.8630
276	0.9274	326	0.9109	376	0.8947	426	0.8786	476	0.8627
277	0.9270	327	0.9106	377	0.8943	427	0.8783	477	0.8624
278	0.9267	328	0.9103	378	0.8940	428	0.8779	478	0.8621
279	0.9264	329	0.9099	379	0.8937	429	0.8776	479	0.8617
280	0.9260	330	0.9096	380	0.8934	430	0.8773	480	0.8614
281	0.9257	331	0.9093	381	0.8930	431	0.8770	481	0.8611
282	0.9254	332	0.9090	382	0.8927	432	0.8767	482	0.8608
283	0.9251	333	0.9086	383	0.8924	433	0.8763	483	0.8605
284	0.9247	334	0.9083	384	0.8921	434	0.8760	484	0.8602
285	0.9244	335	0.9080	385	0.8918	435	0.8757	485	0.8599
286	0.9241	336	0.9077	386	0.8914	436	0.8754	486	0.8595
287	0.9237	337	0.9073	387	0.8911	437	0.8751	487	0.8592
288	0.9234	338	0.9070	388	0.8908	438	0.8747	488	0.8589
289	0.9231	339	0.9067	389	0.8905	439	0.8744	489	0.8586
290	0.9228	340	0.9064	390	0.8901	440	0.8741	490	0.8583
291	0.9224	341	0.9060	391	0.8898	441	0.8738	491	0.8580
292	0.9221	342	0.9057	392	0.8895	442	0.8735	492	0.8577
293	0.9218	343	0.9054	393	0.8892	443	0.8732	493	0.8573
294	0.9214	344	0.9051	394	0.8888	444	0.8728	494	0.8570
295	0.9211	345	0.9047	395	0.8885	445	0.8725	495	0.8567
296	0.9208	346	0.9044	396	0.8882	446	0.8722	496	0.8564
297	0.9204	347	0.9041	397	0.8879	447	0.8719	497	0.8561
298	0.9201	348	0.9038	398	0.8876	448	0.8716	498	0.8558
299	0.9198	349	0.9034	399	0.8872	449	0.8712	499	0.8555

FIGURE 5B.

Standard Abridged Volume Correction Table for Petroleum Oils

GROUP 1 [a]

Legend: t = observed temperature in degrees Fahrenheit; M = multiplier for reducing oil volumes to the basis of 60 F.

t	M	t	M	t	M	t	M	t	M
0	1.0242	50	1.0040	100	0.9841	150	0.9647	200	0.9457
1	1.0238	51	1.0036	101	0.9837	151	0.9643	201	0.9453
2	1.0234	52	1.0032	102	0.9833	152	0.9639	202	0.9449
3	1.0230	53	1.0028	103	0.9830	153	0.9636	203	0.9446
4	1.0226	54	1.0024	104	0.9826	154	0.9632	204	0.9442
5	1.0222	55	1.0020	105	0.9822	155	0.9628	205	0.9438
6	1.0218	56	1.0016	106	0.9818	156	0.9624	206	0.9434
7	1.0214	57	1.0012	107	0.9814	157	0.9620	207	0.9430
8	1.0210	58	1.0008	108	0.9811	158	0.9616	208	0.9427
9	1.0206	59	1.0004	109	0.9807	159	0.9612	209	0.9423
10	1.0202	60	1.0000	110	0.9803	160	0.9608	210	0.9419
11	1.0198	61	0.9996	111	0.9799	161	0.9604	211	0.9415
12	1.0194	62	0.9992	112	0.9795	162	0.9601	212	0.9412
13	1.0189	63	0.9988	113	0.9791	163	0.9597	213	0.9408
14	1.0185	64	0.9984	114	0.9787	164	0.9594	214	0.9405
15	1.0181	65	0.9980	115	0.9783	165	0.9590	215	0.9401
16	1.0177	66	0.9976	116	0.9778	166	0.9586	216	0.9397
17	1.0173	67	0.9972	117	0.9775	167	0.9582	217	0.9393
18	1.0169	68	0.9968	118	0.9771	168	0.9578	218	0.9390
19	1.0165	69	0.9964	119	0.9767	169	0.9574	219	0.9386
20	1.0161	70	0.9960	120	0.9763	170	0.9570	220	0.9382
21	1.0157	71	0.9956	121	0.9759	171	0.9566	221	0.9378
22	1.0153	72	0.9952	122	0.9755	172	0.9562	222	0.9374
23	1.0148	73	0.9948	123	0.9752	173	0.9559	223	0.9371
24	1.0144	74	0.9944	124	0.9748	174	0.9555	224	0.9367
25	1.0140	75	0.9940	125	0.9744	175	0.9551	225	0.9363
26	1.0136	76	0.9936	126	0.9740	176	0.9547	226	0.9359
27	1.0132	77	0.9932	127	0.9736	177	0.9543	227	0.9356
28	1.0128	78	0.9929	128	0.9732	178	0.9540	228	0.9352
29	1.0124	79	0.9925	129	0.9728	179	0.9536	229	0.9349
30	1.0120	80	0.9921	130	0.9724	180	0.9532	230	0.9345
31	1.0116	81	0.9917	131	0.9720	181	0.9528	231	0.9341
32	1.0112	82	0.9913	132	0.9716	182	0.9524	232	0.9337
33	1.0108	83	0.9909	133	0.9713	183	0.9521	233	0.9334
34	1.0104	84	0.9905	134	0.9709	184	0.9517	234	0.9330
35	1.0100	85	0.9901	135	0.9705	185	0.9513	235	0.9326
36	1.0096	86	0.9897	136	0.9701	186	0.9509	236	0.9322
37	1.0092	87	0.9893	137	0.9697	187	0.9505	237	0.9318
38	1.0088	88	0.9889	138	0.9694	188	0.9502	238	0.9315
39	1.0084	89	0.9885	139	0.9690	189	0.9498	239	0.9311
40	1.0080	90	0.9881	140	0.9686	190	0.9494	240	0.9307
41	1.0076	91	0.9877	141	0.9682	191	0.9490	241	0.9303
42	1.0072	92	0.9873	142	0.9678	192	0.9487	242	0.9300
43	1.0068	93	0.9869	143	0.9675	193	0.9483	243	0.9296
44	1.0064	94	0.9865	144	0.9671	194	0.9480	244	0.9293
45	1.0060	95	0.9861	145	0.9667	195	0.9476	245	0.9289
46	1.0056	96	0.9857	146	0.9663	196	0.9472	246	0.9285
47	1.0052	97	0.9853	147	0.9659	197	0.9468	247	0.9281
48	1.0048	98	0.9849	148	0.9655	198	0.9465	248	0.9278
49	1.0044	99	0.9845	149	0.9651	199	0.9461	249	0.9274

Figure 5C.

STANDARD ABRIDGED VOLUME CORRECTION TABLE FOR PETROLEUM OILS

GROUP 2

Legend: t = observed tem. .ature in degrees Fahrenheit; M = multiplier for red. ¿ oil volumes to the basis of 60 F.

t	M	t	M	t	M	t	M	t	M	t	M
0	1.0297	25	1.0174	50	1.0049	75	0.9925	100	0.9802	125	0.9677
1	1.0292	26	1.0169	51	1.0044	76	0.9920	101	0.9797	126	0.9672
2	1.0287	27	1.0164	52	1.0039	77	0.9915	102	0.9792	127	0.9667
3	1.0283	28	1.0159	53	1.0035	78	0.9911	103	0.9786	128	0.9662
4	1.0278	29	1.0154	54	1.0030	79	0.9906	104	0.9781	129	0.9657
5	1.0273	30	1.0149	55	1.0025	80	0.9901	105	0.9776	130	0.9652
6	1.0268	31	1.0144	56	1.0020	81	0.9896	106	0.9771	131	0.9647
7	1.0263	32	1.0139	57	1.0015	82	0.9891	107	0.9766	132	0.9642
8	1.0258	33	1.0134	58	1.0010	83	0.9886	108	0.9762	133	0.9637
9	1.0253	34	1.0129	59	1.0005	84	0.9881	109	0.9757	134	0.9632
10	1.0248	35	1.0124	60	1.0000	85	0.9876	110	0.9752	135	0.9627
11	1.0243	36	1.0119	61	0.9995	86	0.9871	111	0.9747	136	0.9622
12	1.0238	37	1.0114	62	0.9990	87	0.9866	112	0.9742	137	0.9617
13	1.0233	38	1.0109	63	0.9985	88	0.9861	113	0.9737	138	0.9613
14	1.0228	39	1.0104	64	0.9980	89	0.9856	114	0.9732	139	0.9608
15	1.0223	40	1.0099	65	0.9975	90	0.9851	115	0.9727	140	0.9603
16	1.0218	41	1.0094	66	0.9970	91	0.9846	116	0.9722	141	0.9598
17	1.0213	42	1.0089	67	0.9965	92	0.9841	117	0.9717	142	0.9593
18	1.0208	43	1.0084	68	0.9961	93	0.9836	118	0.9712	143	0.9588
19	1.0203	44	1.0079	69	0.9956	94	0.9831	119	0.9707	144	0.9583
20	1.0198	45	1.0074	70	0.9951	95	0.9826	120	0.9702	145	0.9578
21	1.0193	46	1.0069	71	0.9946	96	0.9821	121	0.9697	146	0.9573
22	1.0188	47	1.0064	72	0.9941	97	0.9816	122	0.9692	147	0.9568
23	1.0184	48	1.0059	73	0.9935	98	0.9812	123	0.9687	148	0.9563
24	1.0179	49	1.0054	74	0.9930	99	0.9807	124	0.9682	149	0.9558

STANDARD ABRIDGED VOLUME CORRECTION TABLE FOR PETROLEUM OILS

GROUP 3

Legend: t = observed temperature in degrees Fahrenheit; M = multiplier for reducing oil volumes to the basis of 60 F.

t	M	t	M	t	M	t	M	t	M	t	M
0	1.0361	25	1.0211	50	1.0060	75	0.9909	100	0.9757		
1	1.0355	26	1.0205	51	1.0054	76	0.9903	101	0.9751		
2	1.0349	27	1.0199	52	1.0048	77	0.9897	102	0.9745		
3	1.0343	28	1.0193	53	1.0042	78	0.9891	103	0.9738		
4	1.0337	29	1.0187	54	1.0036	79	0.9885	104	0.9732		
5	1.0331	30	1.0181	55	1.0030	80	0.9879	105	0.9726		
6	1.0325	31	1.0175	56	1.0024	81	0.9873	106	0.9720		
7	1.0319	32	1.0169	57	1.0018	82	0.9867	107	0.9714		
8	1.0313	33	1.0163	58	1.0012	83	0.9860	108	0.9708		
9	1.0307	34	1.0157	59	1.0006	84	0.9854	109	0.9702		
10	1.0301	35	1.0151	60	1.0000	85	0.9848	110	0.9696		
11	1.0295	36	1.0145	61	0.9994	86	0.9842	111	0.9690		
12	1.0289	37	1.0139	62	0.9988	87	0.9836	112	0.9684		
13	1.0283	38	1.0133	63	0.9982	88	0.9830	113	0.9678		
14	1.0277	39	1.0127	64	0.9976	89	0.9824	114	0.9672		
15	1.0271	40	1.0121	65	0.9970	90	0.9818	115	0.9666		
16	1.0265	41	1.0115	66	0.9964	91	0.9812	116	0.9660		
17	1.0259	42	1.0109	67	0.9957	92	0.9806	117	0.9654		
18	1.0253	43	1.0103	68	0.9951	93	0.9800	118	0.9647		
19	1.0247	44	1.0097	69	0.9945	94	0.9794	119	0.9641		
20	1.0241	45	1.0091	70	0.9939	95	0.9788	120	0.9635		
21	1.0235	46	1.0085	71	0.9933	96	0.9782	121	0.9629		
22	1.0229	47	1.0079	72	0.9927	97	0.9776	122	0.9623		
23	1.0223	48	1.0072	73	0.9921	98	0.9769	123	0.9617		
24	1.0217	49	1.0066	74	0.9915	99	0.9763	124	0.9611		

FIGURE 5D.

$\frac{W}{60} \times \frac{.1184}{.9594} = W \times .002057$. If the specified quantity is given directly as G.P.M. it is assumed that it refers to pumping conditions.

(*e*) *Inside Diameter of Pipe* (*d*): Dimensions of pipes to be as given above under General Design. For convenience, Figure 6 is given and shows suitable dimensions for pipe sizes from ¾ inch to 8 inches which covers the range for normal marine design.

TABLE 8

PIPE DIMENSIONS

Size Inches	Actual Outside Diameter Inches	F.O. Booster and Transfer, F.O. Service Pump Suctions. 0 - 200# W.P.		F.O. Service Piping Pump Disch. to Htrs. 201 - 450# W.P.		
		Minimum Thickness Mils	Inside Diameter Inches	Minimum Thickness Mils	Inside Diameter Inches	
3/4	1.050	---	---	95	860	Not included in low pressure curves.
1"	1.315	---	---	95	1.125	
1¼"	1.660	---	---	95	1.470	
1½"	1.900	---	---	95	1.710	
2"	2.375	95	2.185	98	2.179	
2½"	2.875	95	2.685	119	2.637	
3"	3.500	95	3.31	144	3.212	
3½"	4.000	95	3.81	166	3.670	
4	4.500	95	4.31	---	---	Not included in high pressure curves.
4½"	5.000	98	4.808	---	---	
5	5.563	107	5.349	---	---	
6	6.625	127	6.371	---	---	
7	7.625	147	7.331	---	---	
8	8.625	166	8.293	---	---	

FIGURE 6.

(*f*) *Velocity of Flow:* Quantity of oil in G.P.M. may be converted to velocity in feet per second for any size pipe by formula XIII $= \text{Velocity} = V = \frac{.4085 \times G}{d^2}$. The allowable velocity is left to the judgment of the designer.

(*g*) *Equivalent Length of Pipe* (*L*): This is the actual length of pipe plus the equivalent length for fittings, valves, etc., and is the most important and uncertain of any of the factors in the for-

mula. Although the available data is conflicting and there is need for additional experiments, Figures 7, 7A, 7B and 8 have been prepared for the purpose of this article.

Figures 7, 7A and 7B show values of equivalent lengths for various fittings, etc., for turbulent flow as given by several of the articles studied. In using these figures for Marine Design it is to be noted that losses due to entrance, enlargement or contraction

EQUIVALENT LENGTHS (INCLUDES ONLY ITEMS LIABLE TO BE USED ON SHIP) (1)

ITEM		CRANE (W) FROM CHART	NELSON (J) TABLE PAGE 160	McADAMS (K) TABLE PAGE 100	KENT (G)	DARFORTH (G) LIST	DARFORTH (G) CURVE	CRANE, ENGG. (A) TABLES
Stand-	1°	2'-8"	2'-8"	2'-8"		2'-8"		1'-6"
ard 90°	2°	5'-8"	5'-0"	5'-0"		5'-0"	Same as (G) thru	3'-6"
ells	3°	8'-0"	10'-0"	10'-0"		7'-8"	turbulent region;	5'-9"
	4°	11'-0"	13'-0"	15'-0"		10'-0"	increase during	8'-0"
	5°	14'-0"	20'-0"	20'-0"		12'-0"	critical region;	10'-0"
	6°	20'-0"	33'-0"	33'-0"		20'-0"	gradual decrease	13'-0"
							in stream line.	18'-0"
Medium	1°	2'-8"						1'-0"
sweep	2°	4'-8"						2'-3"
ells	3°	6'-8"						3'-6"
	4°	9'-0"						5'-0"
	5°	14'-0"						8'-0"
	6°	18'-0"						11'-3"
Long	1°	1'-8"						0'-9"
sweep	2°	3'-8"						1'-9"
ells	3°	5'-3"						2'-9"
	4°	7'-0"						4'-0"
	5°	11'-0"						6'-6"
	6°	14'-0"						9'-0"
Square	1°	6'-0"		4'-2"				4'-2"
ells	2°	12'-0"		8'-4"				8'-4"
	3°	17'-0"		12'-6"				12'-6"
	4°	22'-0"		16'-8"				16'-8"
	5°	33'-0"		25'-0"				25'-0"
	6°	44'-0"		33'-4"				33'-8"
45 de-	1°	1'-3"						1'-3"
gree	2°	2'-6"						3'-4"
ells	3°	4'-0"						4'-0"
	4°	5'-0"						
	5°	8'-0"						
	6°	10'-0"						
Stand-	1°	6'-0"	5'-0"	5'-0"		5'-0"	Same as (G) thru	3'-8"
ard	2°	12'-0"	10'-0"	10'-0"		10'-0"	turbulent region;	7'-0"
tees	3°	17'-0"	15'-0"	14'-0 "		14'-0"	increase in	11'-6"
flow	4°	22'-0"	20'-0"	20'-0"		20'-0"	critical region;	15'-0"
thru	5°	33'-0"					gradual decrease	22'-6"
branch	6°	44'-0"					in stream line	30'-0"
Run of tee reduced 1/2		Same as standard ells						Same as standard ells
Run of tee reduced 1/4		Same as medium sweep ell						Same as medium sweep ell
Run of standard tees		Same as long sweep ells						Same as long sweep ells

CURVE BASED ON 4" DIAM. STD. PIPE (POTTER)
SP. GRAV 1.0 & 2.0
TURBULENT FLOW LIMITS R = 25 TO 45
CRITICAL REGION R = 60 TO 80
STREAM LINE BELOW R = 80

ABOVE DATA AS CRITICAL, PROVIDED FOR STEAM AND GAS
ABOVE DATA AS TURBULENT, PROVIDED FOR WATER AND VISCOUS LIQUIDS

FIGURE 7.

need not be considered as all tank suctions have or should have belled ends and it is suggested that values for meters, strainers and heaters be obtained from the manufacture of the units used.

EQUIVALENT LENGTHS CONTINUED (2)

ITEM		CHART FF FROM CHART	WELDON (F) TABLE PAGE 100		Hoskins (H) TABLE PAGE 100		over SHEET	DARCY'S (D) LIST	DARCY'S (D) CURVE	CRANE, HOSK. (H) TABLES
Close	1"	6'-0"								2'-3"
return	2"	13'-0"	7'-0"	13'-0"						5'-3"
bends	3"	19'-0"	10'-0"	20'-0"						8'-6"
	4"	25'-0"	13'-0"	26'-0"						12"-0"
	6"	38'-0"	Stream Square line Blank							19'-9"
	8"	50'-0"								28'-9"
Globe	1"	88'-0"	5'-9"		5'-9"			7'-6"	Same as (G) thru	4'-6"
valves	2"	87'-0"	7'-0"		7'-0"			16'-0"	turbulent region;	10'-6"
open	3"	85'-0"	14'-0"		14'-0"			22'-6"	increase in critical	17'-0"
	4"	115'-0"	20'-0"		20'-0"			30'-0"	region; gradual	23'-6"
	6"	170'-0"	30'-0"		30'-0"			46'-0"	decrease in stream	35'-6"
	8"	225'-0"	40'-0"		40'-0"			60'-0"	line.	53'-6"
Angle	1"	15'-0"						3'-4"	Same note as above	2'-6"
valve	2"	28'-0"						6'-8"		4'-9"
open	3"	42'-0"						10'-0"		7'-0"
	4"	56'-0"						13'-4"		10'-6"
	6"	84'-0"						20'-0"		17'-3"
	8"	115'-0"						28'-6"		24'-0"
Gate	1"	0'-7"	1'-0"					1'-0"	Same note as above	0'-8"
valves	2"	1'-2"	2'-0"					1'-10"		1'-4"
fully	3"	1'-8"	3'-0"					2'-8"		2'-0"
open	4"	2'-3"	4'-0"					3'-6"		3'-0"
	6"	3'-6"	6'-0"					5'-0"		5'-0"
	8"	4'-8"						7'-6"		8'-0"
Gate	1"	2'-6"								
valves	2"	7'-0"								
1/4 closed	3"	10'-0"								
	4"	14'-0"								
	6"	20'-0"								
	8"	28'-0"								
Gate	1"	17'-0"								
valves	2"	24'-0"								
1/2 closed	3"	30'-0"								
	4"	38'-0"								
	6"	100'-0"								
	8"	140'-0"								
Gate	1"	70'-0"								
valves	2"	140'-0"								
1/4 closed	3"	200'-0"								
	4"	280'-0"								
	6"	410'-0"								
	8"	550'-0"								

SEE NOTE ON SHEET (1) OF TABULATION

FIGURE 7A.

EQUIVALENT LENGTHS CONTINUED (5).

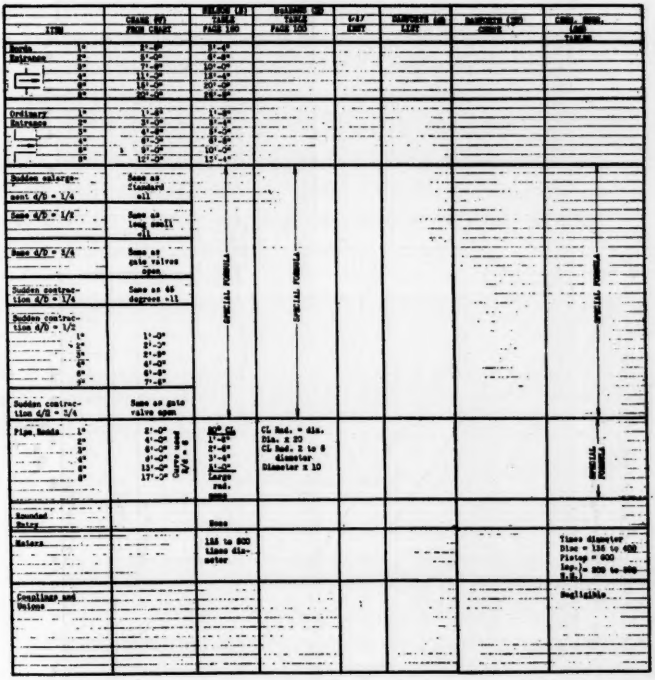


FIGURE 7B.

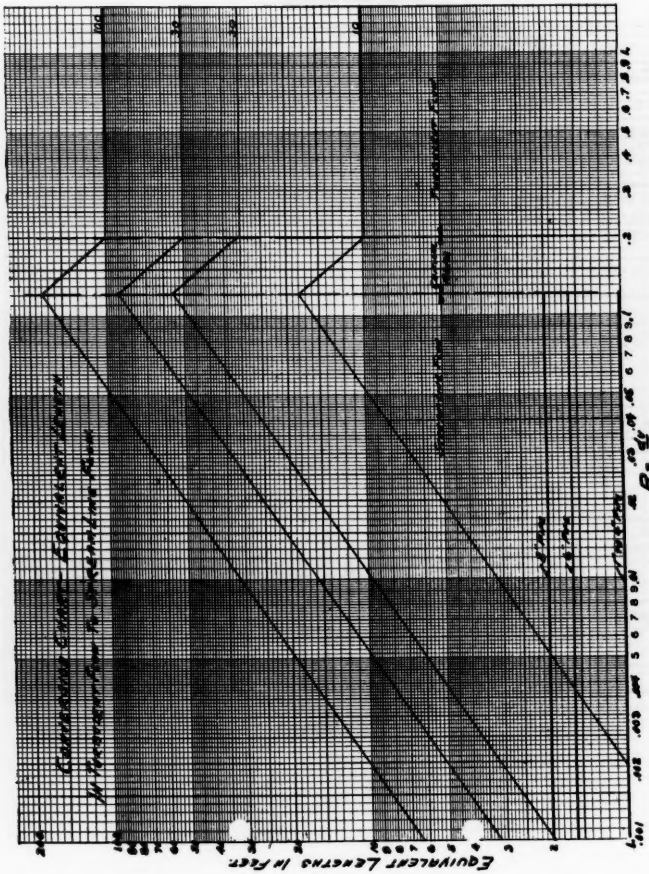

Equivalent Lengths in Feet
Hydraulic Gradient
Coefficient
Velocity
Diameter
Friction
Loss

FIGURE 8.

Figure 8 is a curve from which to derive corresponding values for various values of " R " in stream line flow.

Example: If any particular fitting has an equivalent length of 10 feet in the turbulent range, the corresponding value at an " R " value of .04 would be 8.3 feet. Only four actual values are indicated on the chart (10-30-50 and 100 feet), but other values would be found on lines parallel to these four. This chart may be subject to considerable change, when and if experiments are made for stream line flow.

(*h*) *Friction Pressure Drop* (*P*): Friction pressure drop is given by this formula in pounds per square inch, but may be converted to inches of mercury by the following:

(XIX) One pound per square inch = 2.0416 inches mercury.
It may further be converted to head in feet by

$$(XX)\ \text{Head in Feet} = \frac{P \times 144}{62.3 \times s}.$$

It is to be borne in mind that the friction pressure drop together with static head or lift must not exceed the limits given under General Design.

SAMPLE CALCULATION.

Problem: Basic oil—65 S.S.F. at 122 degrees F.
Basic A.P.I.—8 degrees at 60 degrees F.
Suction pipe to be of sufficient size for 33125 pounds oil per hour when pumped at 700 S.S.F.
Length = 50 feet
Fittings—2 Elbows
2 Tees (straight through flow).
1 Globe valve.

Solution: From viscosity-temperature diagram (Figure 3), the pumping temperature = 65 degrees.

$$\text{Kinematic Viscosity} = y = (2.04 \times 700) - \frac{160}{700}$$

$$= 1428 - .23 = \text{Say } 1428.$$

From Figure 4, 8 degrees A.P.I. = 1.0143 specific gravity at 60 degrees F. = S and equals .1184 gallons per pound at 60 degrees F.

From Figure 5, the multiplier at 65 degrees = .9983.

Then specific gravity at 65 degrees = s = 1.0143

$$\times .9983 = 1.0126, \text{ and G.P.M.} = G = \frac{33125}{60}$$

$$\times \frac{.1184}{.9983} = 65.49.$$

Assume a velocity of 3 feet per second which is equivalent to an inside diameter of 2.96 inches.

Assume from Figure 6 that actual inside diameter will be 3.31 inches (3-inch pipe).

$$\text{Then } V = \frac{.4085 \times 65.49}{(3.31)^2} = 2.44 \text{ feet per second and}$$

$$R = \frac{dV}{y} = \frac{3.31 \times 2.44}{1428} = .00566, \text{ which is far}$$

$$\text{down in the stream line region and } f = \frac{.00207}{R}$$

$$= \frac{.00207}{.00566} = .366.$$

At this low value of R, the equivalent lengths are very small (Figure 8) and are taken as a total of 9 feet.

Then L = 50 + 9 = 59 feet.

$$\text{Then } P = \frac{.323 \times .366 \times 1.0126 \times 2.44 \times 2.44 \times 59}{3.31}$$

= 12.7 pounds per square inch. This is too high for suction pipes as it allows no margin for static lift.

Assume (Figure 6) that a 3½ inch pipe will be used having an inside diameter of 3.81 inches. Then in a manner similar to the above,

$V = 1.84$ feet per second
$R = .00491$
$f = .422$

$$P = \frac{.323 \times 422 \times 1.0126 \times 1.84 \times 1.84 \times 59}{3.81}$$

$= 7.25$ pounds per square inch, which is satisfactory.

Detail Design (Standard Conditions): Instead of using quite complicated curves in an attempt to cover all conditions it is suggested that a series of curve sheets be prepared with each sheet covering one of the standard marine conditions. Each sheet should have an independent curve for each size pipe in the probable range of that particular condition and these curves may be plotted with G.P.M. as abscissa and Pressure Drop in pounds per square inch as ordinates. For convenience it is further suggested that the ordinates be actually in terms of Friction Pressure Drop per 20 feet of equivalent length. If desired, these sheets may have auxiliary scales for converting from pounds per hour to G.P.M. and from drop in pounds per square inch to drop in inches of mercury.

The suggested curve sheets are as follows:

(*a*) Navy grade fuel oil pumped at 700 S.S.F. (65 degrees F.).
(*b*) Navy grade fuel oil pumped at 20 S.S.F. (180 degrees F.).
(*c*) Bunker "C" fuel oil pumped at 700 S.S.F. (105 degrees F.).
(*d*) Bunker "C" fuel oil pumped at 20 S.S.F. (225 degrees F.).
(*e*) Diesel fuel oil pumped at 70 S.S.U. (50 degrees F.).

Due to a possible variation in thickness of pipes, for various standards, no attempt has been made to prepare this entire set of curves. However, "Curve (a)" has been prepared based on pipe dimensions as given in Figure 6 and this sample curve is shown on Figure 9 and the method of preparing this curve follows:

SAMPLE CALCULATION FOR CURVE (*a*) (FIGURE 9).

Kinematic Viscosity $= y = (2.04 \times 700) - \frac{160}{700} = 1428 -$ $.23 =$ Say 1428.

From Figure 4, 8 degrees A.P.I. $= 1.0143$ specific gravity and .1184 gallons per pound at 60 degrees F.

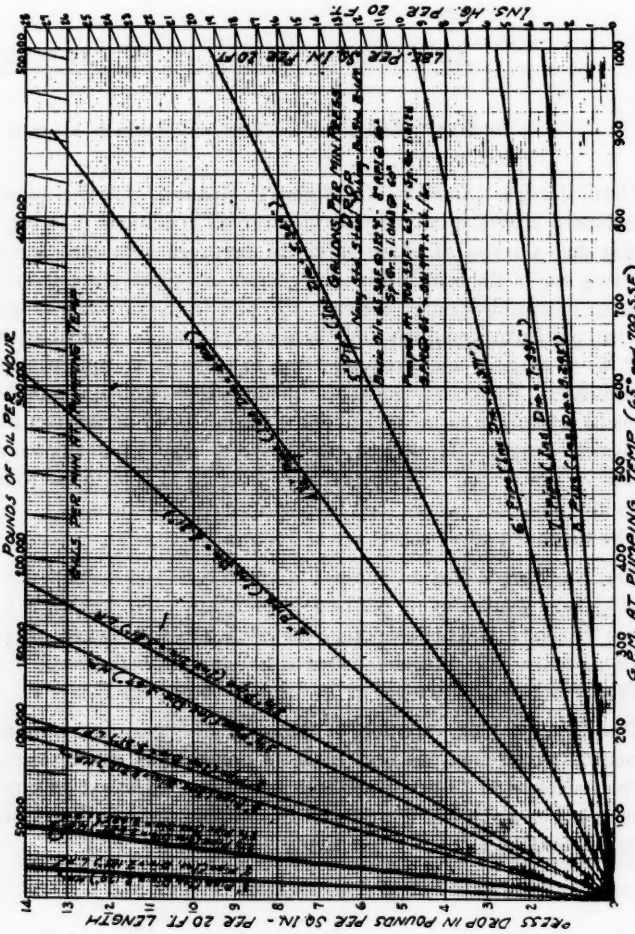

POUNDS OF OIL PER HOUR
G.P.M. AT PUMPING TEMP. (65° and 700 SSF)
PRESS. DROP IN POUNDS PER SQ. IN. PER 20 FT. LENGTH
LBS. PER SQ. IN. PER 20 FT.
INS. Hg. PER 20 FT.

FIGURE 9.

From Figure 5, multiplier at 65 degrees F. = .9983.

Then specific gravity = s = 1.0143 × .9983 = 1.0126.

$$\text{G.P.M.} = G = \frac{.1184}{60 \times .9983} = .001977 \times \text{pounds per hour.}$$

This condition (700 S.S.F.) applies to booster and transfer systems and service pump suction and service pump discharge to fuel oil heaters. Capacity range will probably be from 25 G.P.M. to 1000 G.P.M. for low pressure service and from 25 G.P.M. to 150 G.P.M. for high pressure service (this ignores connections to port and hand service pumps as pressure drop is of relatively little importance in these cases). The range of pipes will probably be 2 inches to 8 inches. Curves are therefore drawn for each size of pipe from 2 inches to 8 inches and scale of G.P.M. is from 0 to 1000.

FIGURE 9.

Assume 8-inch pipe — Diameter = d = 8.293 inches and d^2 = 68.744.

$$\text{Velocity} = \frac{.4085}{68.744} \times \text{G.P.M.} = .00594 \times \text{G.P.M.}$$

$$R = \frac{dV}{y} = \frac{8.293}{1428} \times \text{Velocity} = .00581 \times \text{Velocity.}$$

For stream line flow $f = \frac{.00207}{R}$.

$$P = \frac{.323 \times f \times s \times V^2 \times L}{d} = \frac{.323 \times 1.0126 \times 20}{8.293} \times f \times$$

$V^2 = .7888 \times f \times V^2$ in pounds per square inch per 20 feet of equivalent length.

Assume 500 G.P.M. at 65 degrees F. and the above becomes

Velocity = .00594 × 500 = 2.97 feet per second.

R = .00581 × 2.97 = .0173 (stream line).

$$f = \frac{.00207}{R} \times \frac{.00207}{.0173} = .12.$$

$$P = .7888 \times .12 \times (297)^2 = .835 \text{ pound.}$$

All other points necessary are similarly calculated and plotted as described above. The conversion scale of pounds per hour to G.P.M. at 65 degrees is calculated from G.P.M. = .001977 × pounds per hour and conversion scale from pounds per square inch to inches of mercury is calculated from one pound per square inch = 2.0146 inches of mercury.

If it should be desired to put limiting values on velocity, these curves may be extended only to the G.P.M. falling within these limits.

The curves for condition (*c*) will be similar to those for condition (*a*). The only difference between condition (*a*) and (*c*) is that of specific gravity, which at 105 degrees F. will be 1.0437. Then the pressure drops for this condition will be $\frac{1.0437 - 1.0126}{1.0126}$ = .0307 or 3 per cent higher than values given by curves for condition (*a*).

For condition (*b*) we have

Kinematic Viscosity = 32.8.
G.P.M. = .002057 × pounds per hour.
Specific Gravity = .9731.

This condition (20 S.S.F.) applies only to discharge from heaters to boiler front manifold. Pipe range will be about ¾ inch to 3½ inches high pressure and capacity range about 25 G.P.M. to 150 G.P.M.

Calculations for this curve will be similar to that shown above except that most of the points will fall on the turbulent region and "f" will be taken from Figure 2. Also the curves should be broken so as not to include the critical region between values of R of about .123 to .2.

For condition (*d*) the curves will only vary from condition (*b*) because of specific gravity and will give results 2.84 per cent higher.

For condition (*e*) we have

Kinematic Viscosity = 12.85.
G.P.M. = .0022045 × pounds per hour.
Specific Gravity = s = .9078.

This condition (70 S. S. U.) applies to Diesel fuel oil, both suction and discharge. Pipe range will be about ¾ inch to 3½ inches, and capacity range about 5 to 100 G.P.M. Curve will be similar in type to conditions (*b*) and (*d*).

In conclusion, the author wishes to gratefully acknowledge the valuable suggestions and aid given by Captain C. S. Gillette, U. S. N., and others of the personnel of the Naval Boiler Laboratory, where this survey was made.

LIST OF REFERENCES.

1. Petroleum Refinery Engineering (1936)—W. L. Nelson.
2. Principles of Chemical Engineering (1923)—Walker, Lewis & McAdams.
3. Fuel Oil Viscosity-Temperature Diagram—Form N. Eng. 264.
4. Heat Transmission (1933)—Wm. H. McAdams.
5. Friction Pressure Loss in Oil Lines (1920)—Kinney Mfg. Co.
6. Oil Flow in Pipe Lines (Second Edition, 1927)—R. S. Danforth.
7. Oil Flow—Viscosity and Heat Transfer (1923)—R. S. Danforth.
8. Flow of Oil in Pipes (April, 1922)—"Lubrication."
9. National Standard Petroleum Oil Tables—Bureau of Standards No. 154.
10. A.S.T.M. Standards on Petroleum Products and Lubricants (Sept., 1935).
11. Engineering Data on Flow of Fluids in pipes (1935)—Crane Co.
12. Kents Mechanical Engineering Hand Book—II Power (1936).
13. Chemical Engineers Hand Book (1934)—J. H. Perry.
14. A study of the Data on the Flow of Fluids in Pipes (A.S.M.E., 1932)—Emory Kemler.
15. Flow in Pipes (A.S.M.E., 1928)—M. D. Aisenstein.
16. The Flow of Fluids in Closed Conduits (A.S.M.E., 1932)—R. J. S. Pigott.
17. Nomographic Charts for Determining Fluid Friction in Pipe Lines ("Refiner," 1936)—H. V. Beck.

FOUNDATIONS FOR AUXILIARY MACHINERY.

By Lieutenant Commander Leonard Kaplan, (CC), U.S.N., Member.*

As the author points out in this article, it is surprising how varied the practice in foundationing auxiliary machinery on board ship is found to be, differing as it does between shipbuilding concerns, and even between auxiliaries on the same ship. He starts by stating a few sound premises upon which all can agree, and from these develops a standardized approach which may be followed with the certainty of some improvement in average shipbuilding practice. He discusses the choices of method available, eliminates the most undesirable, and throws much light on a little understood and little discussed shipbuilding problem. His discussion should be valuable in the drafting rooms of all shipbuilders, and of their sub-contractors as well.

FOREWORD.

Practice varies widely in the types of foundations provided for the installation of auxiliary machinery in naval vessels. There is a lack of uniformity also in the design of bases or bedplates of the mechanical units supplied by the different makers of auxiliary equipment. Certain of these foundations and bedplates, in the writer's opinion, are definitely objectionable and militate against proper "lining up" of the machinery aboard ship.

There is nothing novel in the foundation designs put forward in this paper as indicative of the "best practice." On the contrary, the material assembled here represents rather an effort to sort out what is good from what is not so good in existing types of foundations and bedplates. The discussion presented is offered more in the hope that it will stimulate thought on a subject that heretofore has not been given the study it merits, than with the idea of win-

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ning converts to a specific method of securing auxiliary machinery in ships. Were it possible for shipbuilders generally to agree on what constitutes a satisfactory foundation for, say a motor driven pump set, the manufacturer's problem of furnishing a pump set mounted on a bedplate suitable for attachment to this "standard" foundation would be greatly simplified.

It is with this in mind, and assuming that the ideas advanced will be judged in the light of the precept enunciated in the foregoing paragraph, that the following "tentative standards" are proposed.

GENERAL PRINCIPLES.

Two basic principles of design may be formulated at the outset. Stated negatively—and it has seemed more effective to do this because they are stipulations more frequently disregarded than observed—, they are:

First—Do not bolt through the deck, which is tantamount to saying that the foundation and the holding down bolts should be kept above the deck.

Second—Use no canvas or other packing between faying surfaces, and no grommets under the heads of bolts.

The second proposition is in fact a corollary of the first, for if the foundation bolts are kept above the deck, it will not be necessary to use packing between faying surfaces or grommets under the heads of bolts in order to obtain water-tightness.

The author cannot emphasize too strongly the importance of these two design premises. They are the primary objectives, as he sees it, of every foundation job, large or small. The details of construction elaborated in the following pages all converge upon the realization of these desiderata.

DETAIL CONSIDERATIONS.

(*a*) Take first the simplest type of foundation, where the machine to be mounted is carried on legs having relatively small areas of contact. In this class will be found such items as lathes, drill presses, and other machine tools; laundry appliances; galley equipment; etc. They are best secured with a thick block under each leg, welded to the deck, and tapped to receive a stud. The bolting is done with a nut and lock nut, after levelling up, as shown in Figure 1 below.

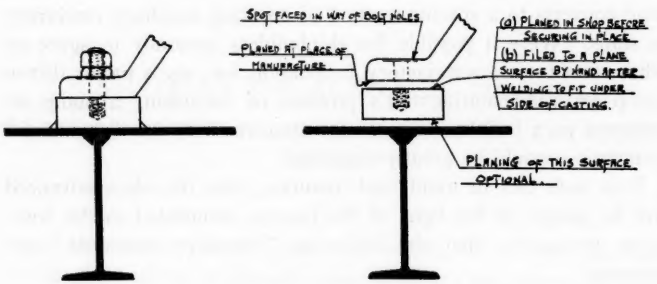

SPOT FACED IN WAY OF BOLT HOLES
PLANED AT PLACE OF MANUFACTURE
(a) PLANED IN SHOP BEFORE SECURING IN PLACE
(b) FILED TO A PLANE SURFACE BY HAND AFTER WELDING TO FIT UNDER SIDE OF CASTING.
PLANING OF THIS SURFACE OPTIONAL.

Fig. 1

A foundation of this kind has several advantages. First of all, it satisfies the requirement of not penetrating the deck. Secondly, it makes it possible to get the holding down bolt (stud and nut combination in this instance) directly over the deck stiffener. In addition to these, it is simple and inexpensive.

If the block is tapped for its whole thickness, care must be exercised not to allow the stud to act as a screw jack tending to back off the block by putting the welds that secure it to the deck in tension.

It is to be assumed that the contact surfaces of the legs of the machine will have been planed by the manufacturer. The blocks to be welded to the deck should likewise be planed, top and bottom, before they are brought down to the ship. Strictly speaking, it is not necessary to take a cut on the bottom surface, which will at best establish only an approximate bearing with the irregular surface of the deck plating. But the top surface should be planed, because even if it has to be shaped to a different plane to fit the under side of the leg of the machine, the work of fitting it will be simplified by having a plane surface to work from. It is expected that the actual lining up here will be accomplished by hand filing the top surfaces of the welded blocks to fit the individual legs of the machine.

The top of the legs should be spot-faced in way of the holes drilled for the holding down bolts, in order that the nut immediately above will establish accurate contact (bearing) in a plane normal to the axis of the bolt.

The nut should be locked with a jam nut, and the thicker nut should always be placed nearer the threaded (or free) end of the bolt. The reason for the latter requirement is that the strain is carried in the threads of the nut nearer the end of the bolt—the upper nut in the case illustrated—because this is the last one to be set up.

The double nut combination is always to be preferred to a single nut and lock washer. While lock washers (also called spring washers) have extensive application in commercial work, the wide variety and lack of uniformity of the types offered, the chance of breaking them if they are made of material that is too hard, or of compressing them, if the material is too soft, to such an extent that they no longer supply the spring tension and frictional resistance to turning of the nut, which they are designed to effect, all combine to recommend against their use.

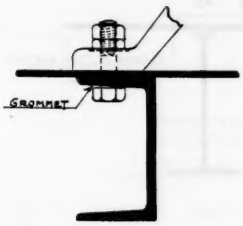


Fig. 2
NOT RECOMMENDED

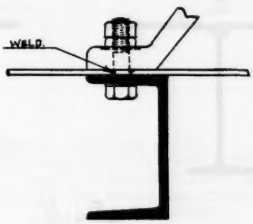


Fig. 3
NOT RECOMMENDED

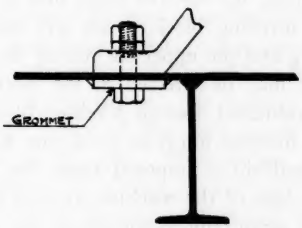


Fig. 4
NOT RECOMMENDED.

It seems hardly necessary to say that tap bolts have no place in a foundation job. Where through bolting is not possible or is impracticable, preference should always be given to studs and nuts.

Figures 2, 3 and 4 show other foundation details, of more or less frequent occurrence, for securing the types of auxiliary machinery falling within this category. The superiority of the foundation illustrated in Figure 1 should be obvious.

The argument will be offered at times that there is an advantage in inserting an extra liner or levelling block between the welded pad and the leg, as shown in Figure 5, to facilitate the work of lining up.

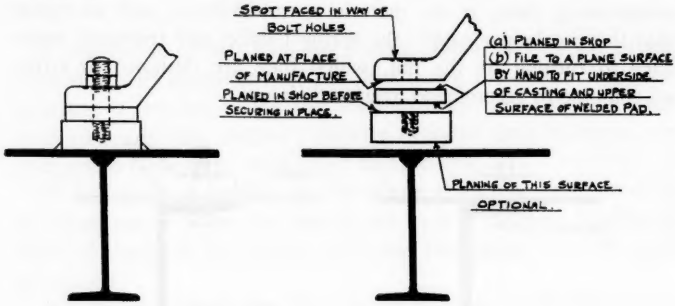

SPOT FACED IN WAY OF BOLT HOLES
PLANED AT PLACE OF MANUFACTURE
PLANED IN SHOP BEFORE SECURING IN PLACE.
(a) PLANED IN SHOP
(b) FILE TO A PLANE SURFACE BY HAND TO FIT UNDERSIDE OF CASTING AND UPPER SURFACE OF WELDED PAD.
PLANING OF THIS SURFACE OPTIONAL.

Fig. 5

If the top surface of the pad has not been distorted by welding —and this would appear to be a pretty good assumption, for it is not likely that the surface of a thick block will be deflected very much from a true plane by the heat incident to attaching it to the deck—, the work of lining up resolves itself into developing two plane surfaces on the levelling block which will match those of the under side of the leg and the upper surface of the pad. While some economy of time may be effected in the work of installation by the use of an additional liner or levelling block, the writer does not agree that the finished job is as good, nor does he believe that any very great hardship is imposed upon the machinist by requiring him to fit the legs of the machine directly to the welded pads. Additional liners serve only to complicate the assembly and to build up a needless instability factor in the completed installation.

(*b*) Consider next what for lack of a better name may be called the intermediate type of foundation. It is applicable to the large group of auxiliaries that require something more than a welded pad foundation, yet do not need continuous support over the entire area of their bases. Included in this group are self-contained high speed units such as motor driven oil and water pump sets, air compressors, motor generators, and the like, none of which transmit any great amount of unbalanced thrust to their foundations. The mechanical auxiliaries falling within this category are generally carried on a framework of channel shaped beams, and the final alignment of the units themselves with respect to these supporting shapes will in most instances have been accomplished at the place of manufacture. The problem of installation is therefore one of providing adequate support from the ship's structure and a practical means for levelling up.

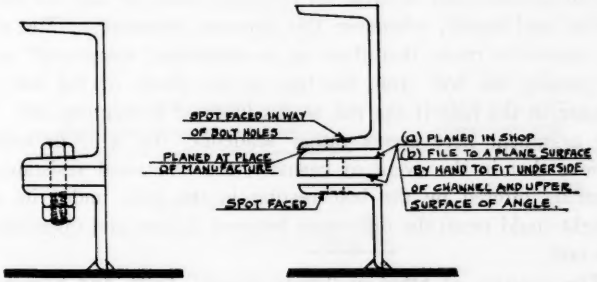

SPOT FACED IN WAY OF BOLT HOLES
PLANED AT PLACE OF MANUFACTURE
SPOT FACED
(a) PLANED IN SHOP
(b) FILE TO A PLANE SURFACE BY HAND TO FIT UNDERSIDE OF CHANNEL AND UPPER SURFACE OF ANGLE.

Fig. 6

Figure 6 shows one method of accomplishing this. The attachment is made to an angle welded toe to the deck, and the channel-shaped skids of the unit to be installed are bolted to these, with levelling blocks in way of each bolt inserted between.

The horizontal flange of the angle should be planed before it is brought down to the ship. This surface should remain a true plane, even after it is welded, if the angle is of rugged section. The bottom flange of the channel bedplate, it may be assumed, will have been planed by the manufacturer. The job of lining up therefore consists only of working two plane surfaces on the

levelling blocks (or spacers). The blocks should not be too large, and the hole for the bolt should pass through the geometric center of the block, if practicable, after the manner of a rectangular washer. If this is done, there will be less tendency for the block to work loose than if the bolt is put down through a corner of the block. The number and the spacing of the bolts and blocks will, of course, depend on the magnitude of the load to be carried.

Through bolts are used, and these are secured with nut and lock nut, after the manner described in (*a*) for the welded pad foundation. In most cases, it will be necessary to pass the bolt from the top, because the angle will not be deep enough to permit passing it from the bottom. Where a choice exists, the writer's preference is for passing the bolt up from the bottom and putting the two nuts on top. This preference is based mainly upon considerations of accessibility for setting up on the nuts when making the installation, and likewise for breaking down the unit for examination and repair, whenever this becomes necessary. The claim is sometimes made that there is an additional margin of safety in passing the bolt from the top, as the shank of the bolt will remain in the hole if the nut works loose or is snapped off. But the argument here seems rather academic, for it is difficult to believe that in the event of casualty, the additional resistance to shearing offered by the bolt resting in the hole under its own weight could mean the difference between failure and operation of the unit.

The question of fitted or "body bound" bolts, and how many should be specified for a particular installation, is one which should be decided on the individual merits of each case.

Figure 7 shows a typical layout for a heavy motor-driven oil pump set. The pump is rated 281.25 gallons per minute at 540 pounds pressure; and is driven through a 4.1 : 1 reduction gear by a 440-volt, 3 phase, 115 H.P. motor running at 1755 R.P.M.

The upper side of the connecting flange of the channel should be spot-faced in way of each holding-down bolt, to provide a bearing surface for the under side of the bolt head at right angles to the axis of the bolt. But it will be necessary to remove too much of the metal in the flange, with consequent impairment of the strength of the section, unless channels are chosen which have nearly parallel sided flanges. An interesting study is given below

(Figure 8) of the differences in shape and structural values of an 8-inch deep, 18.75 pound channel of the "American Standard Channel" series with one of the "Shipbuilding Channel" series of the same depth and very nearly (18.7 pounds) the same weight.

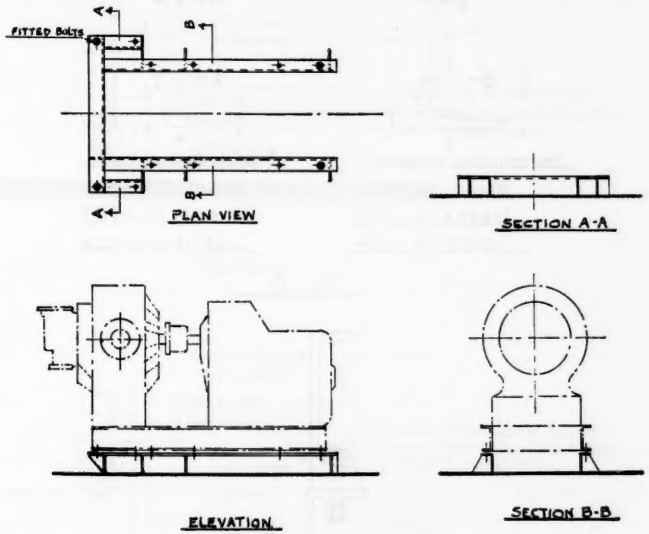


Fig. 7

It is at once apparent that the shipbuilding channel, pound for pound, is the more efficient structural shape. The greater width of flange, 3 inches for the shipbuilding channel as against 2½ inches for the standard channel, also contributes greatly to the convenience of assembly, since it provides more room for the head of the bolt, the gage line of which is moved out to 1¾ inches from the heel of the bar instead of 1½ inches, and so eliminates the necessity for crowding the bolt head up in the bosom of the beam.

Where shapes of the standard series are used (shapes, that is, having a pronounced lack of parallelism of the sides of their flanges), the alternative is to restore this parallelism by welding in tapered wedges after the manner shown in Figure 9, the upper surfaces of the wedges in turn being spotfaced.

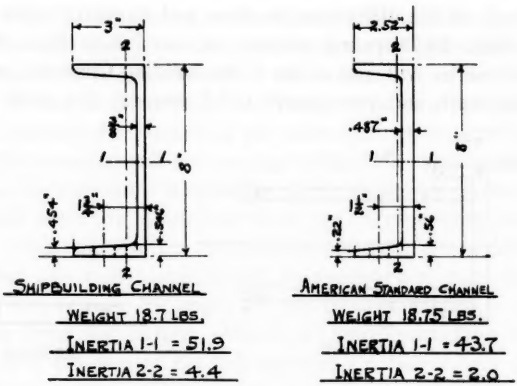


Fig. 8

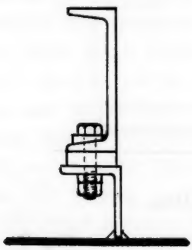


Fig. 9
NOT RECOMMENDED.

This would appear to be a poor substitute for a shipbuilding channel with spot-facing of negligible depth in way of the bolt hole. Here, too, it is important to remember that the fewer pieces a job is made in, the better the resultant assembly.

(*c*) Finally, there is the heavy duty type of foundation, where bearing support is provided over the whole contact area of the base of the machine, usually a casting. The largest and most important auxiliaries belong to this class. Size in itself is not the primary distinguishing characteristic, however, but the fact that these mechanisms do carry an appreciable thrust or reaction into

their foundations. Windlass, anchor engines, boat crane machinery, and secondary gun mounts may be cited as illustrative of the kinds of auxiliaries that require foundations of the most rugged construction.

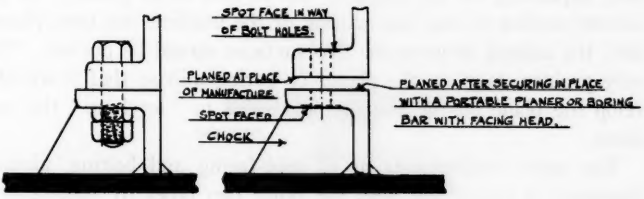

SPOT FACE IN WAY OF BOLT HOLES.
PLANED AT PLACE OF MANUFACTURE.
SPOT FACED
CHOCK
PLANED AFTER SECURING IN PLACE WITH A PORTABLE PLANER OR BORING BAR WITH FACING HEAD.

Fig. 10

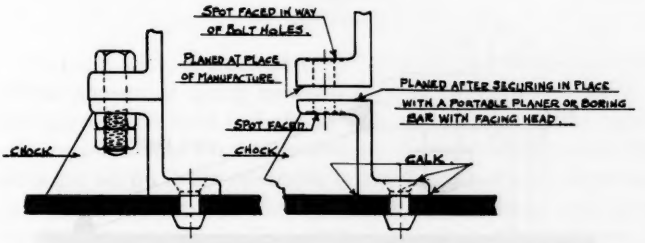

SPOT FACED IN WAY OF BOLT HOLES.
PLANED AT PLACE OF MANUFACTURE.
SPOT FACED
CHOCK
CHOCK
CALK.
PLANED AFTER SECURING IN PLACE WITH A PORTABLE PLANER OR BORING BAR WITH FACING HEAD.

Fig. 11

Figures 10 and 11 show two such foundations. The essential difference between them is that one is welded and the other riveted. In most ordinary cases, the welded job would be acceptable, but in way of special treatment steel, or wherever welding for any other reasons might be objectionable, the riveted foundation should prove equally satisfactory. The riveted flange of the bar should be fitted as neatly as possible, calked toe and heel; the rivets countersunk, driven from the top, and calked as shown. Properly done, it should not be necessary here to use packing between the faying surfaces, or to "gun" the bar afterwards.

If the job is important enough to warrant a foundation providing continuous bearing support, planing of the surface in the ship, after the bar has been welded or riveted, would appear to be justified. This can be accomplished with a portable planer or boring mill, depending on the shape of the surface to be planed. If the bottom surface of the base casting of the machine has been planed also, the contact between the two surfaces should be perfect. The only problem is to set the planer or boring mill so that it will develop the plane on the foundation needed to "level up" the machine.

The same requirements as to spot-facing and bolting, already discussed in connection with the other two types of foundations, apply with equal effect in this case.

Still another variation of the heavy duty foundation may be seen in Figure 12 below, where a built-up shape is used in lieu of an angle.

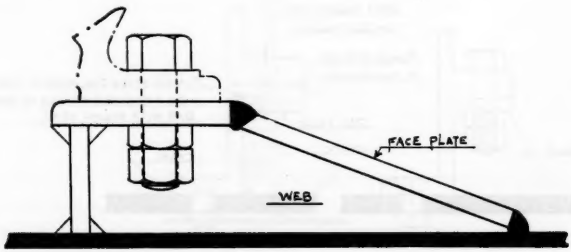


Fig. 12

Here, too, it may be said as in a previous paragraph, that it is better to use a rolled shape of the proper size, if one can be found or cut from a larger shape, reasoning that one piece is better than two or three.

More instances can be found probably in current shipbuilding practice of foundations which do not conform, than those which do conform, with the principles laid down in this part of the discussion relating to the installation of heavy duty auxiliary machinery. It seems well for this reason to review in some detail the objections to the more conventional type.

Consider first the foundation shown in Figure 13 below, where a solid liner has been fitted under the base of the machine, and the whole assembly bolted to the deck and through it, and the headers underneath as well.

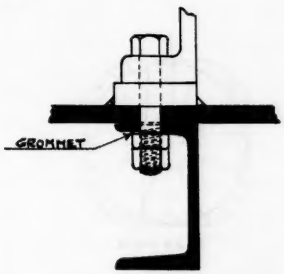
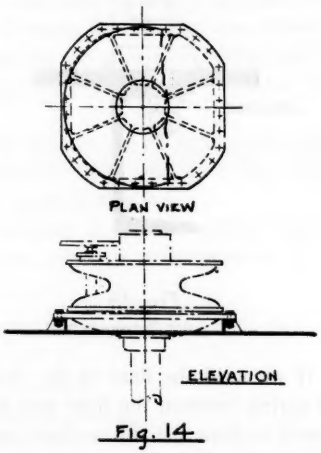


Fig. 13
NOT RECOMMENDED.

To begin with, if we weld the liner to the deck, we lock up a certain amount of spring between the liner and the deck, for it is the nature of the weld to draw the two surfaces apart. The apparent compressibility of the thicknesses of the assembly when the bolts are set up is not conducive to the maintenance of alignment. If we rivet the liner to the deck, or even rivet first and weld afterwards to insure that the liner or doubler is "up," the tightness of the quilting rivets will be a matter of some concern, for it may be necessary to plane away most of the countersinks, when the final cuts are taken. The rivets can of course be recalked, without material loss in the total amount of bearing surface. Grommets must be used to insure the tightness of the bolts. Even so, the faying surface between the header and the deck is open, and so also is the plane of bearing between the base of the casting and the solid liner.

The only advantages gained are (1) better headroom in the machinery space, and (2) a neater tie-up of headers with the beams under the deck. It is the author's opinion that these are greatly outweighed by the unfavorable factors enumerated in the foregoing paragraph, and also by the lack of accessibility for care and preservation.

Illustrative of the idea of keeping the supporting structure and the bolts above the deck, is the outline drawing, Figure 14, of the foundation for a windlass. Continuous bearing for the base of the wildcat is provided by a heavy angle framework, welded toe to the deck.



FLEXIBLE FOUNDATIONS.

For the sake of completeness, a word is included here about flexible foundations. What has been written up to this point has been concerned with furnishing a rigid attachment to the ship's structure—rigid, that is, within the limit of elasticity of the material used. A rigid foundation will, of course, carry into the machine secured to it, hull vibrations set up by other machines; or, for that matter, hull vibrations from any source. The writer's only experience with flexible foundations is confined to ventilation sets, mounted on rubber-inserted vibration dampeners, after the manner shown in the accompanying sketch, Figure 15.

Intake and delivery connections are made by means of canvas joints, so that the blower is free to move with reference to the ship's structure. Service reports indicate that the vibration dampeners have not worked too well, the trouble being that the vulcanizing bond of rubber to steel fails to hold.

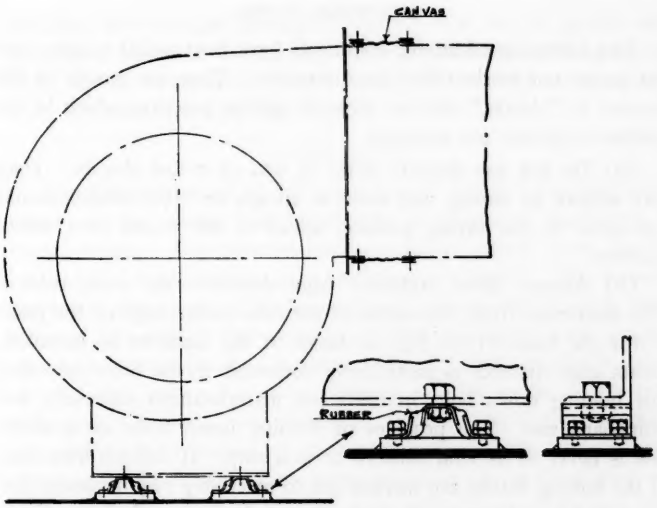


Fig. 15

Whether the principle of the flexible foundation is sound or not, the author is not prepared to say. Some would extend its application to a wide variety of self-contained units having no great amount of unbalanced thrust within themselves, and carrying negligible reactions therefore into their foundations. Take for example a precision grinder installed in the machine shop of a high speed vessel. Would it be better to introduce more flexibility (one way of doing this would be to use lighter angles with longer legs) into the foundation, so that the vibrations of the ship's structure due to other causes would not be carried into the grinder, or go after the problem by building a still heavier and yet more rigid foundation under it? The latter method, it must be admitted, is at best a palliative, because it does not strike at the cause of the trouble—merely seeks instead to reduce the amplitude of the vibrations. Yet the flexible foundation might only succeed in magnifying the effect of the hull vibrations. It may be that the correct answer to the question is that a precision grinder has no place on a high speed vessel of light construction.

ADDITIONAL NOTES.

The following additional comments have been added to take care of points not worked into the discussion. They are mostly in the nature of "don'ts," and are directed against practices which in the author's opinion are unsound.

(*a*) Do not use flanged plates in lieu of rolled shapes. They are seldom as strong, and there is always an objectionable round or curve to the faying surface, which is not found in a rolled section.

(*b*) Always allow sufficient edge distance—the rivet rule is 1⅝ diameters from the center of the hole to the edge of the plate —for the holes in the legs or bases of the units to be installed. Scant edge distance is particularly noticeable in the holes provided for holding down bolts in units not manufactured especially for shipboard use. The problem of holding down bolts on a shore job is never of as vital concern as in a ship. It follows then that if the bolting details are worked out to meet ship requirements, the same design will answer for both shore and ship installations. Each hole should be located, too, so that the bolt has a chance to develop the strength of the leg. If this cannot be accomplished with a single bolt, a pattern for two or three bolts, or whatever number is required, should be chosen, and the holes for them drilled in the leg of the machine.

(*c*) Details of the method of bolting and locking should be clearly delineated in the shipbuilders' plans. There is a tendency too often to cover this part of the drawing in "Notes," and to leave much of the actual design work to the outside personnel. Explanation for this delinquency may be found in the fact that sub-contractors' drawings are often rather hazy on bolt holes and other foundation details, and it is therefore not possible for the shipbuilders to get the information into their plans at the time they are prepared.

(*d*) Under "Detail Considerations (c)," planing of the surface of the heavy duty type of foundation has been stated as a requirement. The writer has in mind that this is not always the procedure followed; that in many cases, the machinists will set to work with an aerial grinder and attempt to develop a plane surface by this means. It is extremely unlikely that this method will

establish the intimate bearing or contact that can be gained with a portable planer or boring mill. Investment in the planing equipment should be repaid in a short time by the savings in labor costs over the hand grinding job.

CONCLUSION.

The ideas presented in this article are in effect a compilation of notes on the design and installation of auxiliary machinery in naval vessels. The material has been drawn largely from the experience of four years on new warship construction. The author's principal aim, as stated in the foreword, has been to promote some interest in a subject for which no definite outline of good practice has been promulgated. On some points brought out in the monograph, the writer has very definite opinions—the "American Standard Channels" for example, which he feels have no place in a bolted (or riveted) job of any kind aboard ship. At the other extreme is the flexible foundation, which has been interjected into the discussion more as a poser with interesting possibilites, than with the thought of contributing anything to the available information on that subject. In the main, however, the ideas put forward are not part of a doctrine or dogma, but are indicative only of what the author considers to be the best in current practice.

THE EFFECT OF COLD WORK ON THE CORRODIBILITY OF CAST CORROSION RESISTANT STEEL IN SALT SPRAY AND NITRIC ACID.

By Joseph A. Duma.*

As those readers of the Journal *interested in metallurgy will recall, Mr. Duma contributed an article in the August, 1936, issue of the* Journal *which dealt with some peculiar properties of 70-30 Copper Nickel Condenser Tubes which had been purchased under the then existent specifications. Mr. Duma's article created some discussion at that time and the careful laboratory work of the Norfolk Navy Yard subsequently caused some revision of the specifications. In this discussion, Mr. Duma presents another original investigation of his laboratory which should be interesting in view of the rather widespread belief that the cold working of cast corrosion resistant steel is deleterious to its corrosion resisting properties. As he points out, only steel produced under the technique followed by the Norfolk Navy Yard was examined, but the results obtained certainly seem to indicate that for this steel, at least, cold working of itself does not necessarily reduce the corrosion resistant properties.*

Naval engineers and designers are continually being informed of the corrosion idiosyncrasies of corrosion resisting steel in a marine environment. They have heard, and know the meaning of, intergranular corrosion and chromium impoverished grain boundaries, of anodic pitting and oxygen concentration cells, of weld decay and sensitization; they also know what it is to passivate and to stabilize corrosion resisting steel. These and a multitude of other terms, as applied to corrosion phenomena, are familiar to them.

* Assistant Metallurgist, Industrial Department, Navy Yard, Norfolk, Va.

The precise effects of cold work, however, upon the corrodibility of that metal in an atmosphere charged to saturation with finely divided salt spray are but partially known. The author has tested the resistance to salt spray of many cold-worked specimens, namely: Ground or filed spot welds; chipped, peened, and then ground beads of deposited weld metal; bent and subsequently straightened sheet; cold-sheared sheet; small sandblasted parts; etc. And always prior to testing, the cold-worked specimens were carefully washed in the following cleaning agents, in this order: Petrolic ether, alcohol, 50 per cent cold nitric acid (immersion for 30 minutes), warm water, and alcohol. The purpose of the nitric acid treatment was to dissolve embedded and adhering steel grit as well as other foreign impurities lodged in the metal's surface by worn or dirty tools.

Since the resistance to rusting of the above specimens was consistently lower than that of standard test specimens made from cast corrosion resisting steel of the Yard's manufacture, it was believed that cold work possibly adversely affected this property. The effect of cold work was investigated as follows:

ANALYSES OF STEELS TESTED.

In Table I there are listed the analyses of twelve different heats of Grade 1, corrosion resisting steel. These heats are not laboratory, or experimental tests, but represent actual commercial castings produced in a basic-lined electric arc furnace in melts of 1000 to 2000 pounds each. Inasmuch as the Yard's practice for manu-

TABLE I

Chemical Composition of Test Specimens

Specimen Number	C.	Mn.	P.	S.	Si.	Ni.	Cr.	Ti.
1	.14	.61	.016	.010	.24	10.83	22.49	.20
2	.11	.64	.028	.010	.83	10.40	21.56	.22
3	.14	.60	.024	.017	.27	11.60	22.34	.19
4	.10	.74	.029	.006	.79	11.40	23.91	.46
5	.13	.68	.009	.004	.95	11.33	23.32	.33
6	.14	.61	.016	.010	.24	10.83	22.49	.20
7	.03	.58	.014	.020	.27	9.88	22.88	.17
8	.08	.60	.012	.014	.67	10.36	21.08	.18
9	.12	.62	.013	.008	.45	10.98	23.02	.36
10	.12	.56	.019	.009	.23	11.65	22.68	.38
11	.12	.56	.019	.009	.23	11.65	22.68	.38
13	.13	.60	.022	.006	.77	11.11	21.66	.22
14	.13	.70	.024	.008	.73	10.90	21.75	.36
15	.12	.56	.019	.009	.23	11.65	22.68	.38

facturing corrosion resisting steel is unique, differing from conventional methods with respect to the type of slags employed and the introduction of titanium, a brief description of the process is included.

One thousand pounds of stainless steel scrap, together with 500 pounds foundry returns, plus a small quantity of both nickel and iron oxide, and 10-15 pounds lime comprise the initial charge of an average heat. When the charge is completely melted the resulting lime slag is thinned down with 10-12 pounds alumina. This thin, grey slag is allowed to remain in the furnace from 5 to 10 minutes; it is then cleaned off and replaced with one consisting, when first made, of 10-12 shovelfuls of low silica lime. From time to time, in the course of refining, this newly made slag is occasionally sprinkled over with half shovelfuls of a 1:10 aluminum shotlime mixture. At this stage approximately 40 pounds of 70 per cent ferrochromium, together with about 4 pounds of 95 per cent ferrosilicon are incorporated into the bath. This addition is not made at one time but split in two and introduced to the bath five minutes apart. At the end of the final chromium addition the furnace is tilted, and some of the second slag is run off. The final slag is repaired with several shovelfuls of aluminum-lime. The bath is then ready for the final alloy addition; i.e., titanium, which is added to the bath in the form of a mixture made up of 2 parts rutile and 1 part shot aluminum. An intense exothermic reaction ensues between the rutile (titanium dioxide) and the aluminum, resulting in their simultaneous reduction and oxidation. At the completion of the thermite reaction the metal is ready to pour.

SALT SPRAY TEST AND SPECIMENS.

Dimensions—The specimens for salt-spray test were of the standard plumb bob type, machined to the form and dimensions shown in recent Navy Department Specifications for Corrosion Resisting Steel.

Salt-spray Test—The salt-spray test was conducted by exposing the specimens prepared as above to a finely divided spray generated by a hard rubber atomizer using air at 15 pounds pressure and a 4 per cent sea-salt solution. The chemical composition and alkalinity of the salt used were:

Sodium chloride (NaCl), per cent	98.78
Insoluble in water, per cent	1.01
Lime (CaO), per cent	0.10
Magnesia (MgO)	Trace
Oxide of iron (Fe_2O_3)	None
Moisture or loss at 105 degrees C., per cent	0.10
pH value of 4 per cent solution in tap water	7.4 – 7.5
pH value of 4 per cent solution in distilled water	7.4

To attain maximum uniformity of exposure, every specimen was periodically rotated 180 degrees so as to face the fog and all specimens were alternately transposed from one rack to the other. The average prevailing test temperature was 25 degrees C. (77 degrees F.).

NITRIC ACID TEST SPECIMENS.

Dimensions—The specimens for nitric acid test were 2 × ½ × ¼ inches.

Nitric acid test—The test was conducted by immersing specimens, two at a time, in 400 ml of C.P. 67 per cent concentrated nitric acid (s. g. 1.42) contained in flasks equipped with reflex condenser. Dilution of the 67 per cent acid solution was accomplished by using distilled water. The flasks were heated until boiling occurred and were maintained at boiling temperature for forty-eight (48) hours after which the samples were withdrawn, washed, dried, and weighed. Above procedure was repeated three times on each pair of specimens.

The corrosion losses are expressed in inches penetration per month and are calculated after each 48-hour period from the formula,—

$$R = \frac{43.\ 9\ W}{A\ S\ T}$$

where

R = Corrosion rate in inches penetration per month.
W = Weight loss in grams.
A = Total area in square inches.
S = Density in grams per ml.
T = Time in hours (48).

TREATMENT, COLD WORK, SURFACE FINISH.

All the test specimens used in the present investigation were made from corrosion resisting cast steel coupons, 9 × 1½ × 2 inches, quenched in cold water from 2000 degrees F., after having been held for two hours at the stated temperature.

In order to qualify for the present tests, each specimen of the 15 used was given a preliminary acceptance test consisting of 100 continuous hours of exposure to the action of 4 per cent sea salt fog. This test served to pass the castings made from the heats which each of the specimens represented. The finish on all the preliminary test specimens was standard; i.e., it was obtained by using dry No. 00 iron-free, new emery cloth.

After having successfully passed the preliminary test, the specimens were subjected to varying degrees of cold work. As shown in Table II, the first 4 specimens were strain-hardened on their surface 3 to 10 Rockwell "B" points by sandblasting. Specimens 5 and 6 were hand-hammered over their conical and cylindrical surfaces to still higher values of hardness. Specimens 7, 8 and 9 were work-hardened by "cloudbursting." This treatment, when

TABLE II- Work Hardness Data

Specimen Number	Nature and Extent of Cold Work	Work Hardened From	To	Increase
		Rockwell "B"		
1	Sand blasted - 2 mins; 90 lbs./sq.in. pressure	87-89	90-92	3
2	" " - 4 mins; " " " "	82-84	88-90	6
3	" " - 6 mins; " " " "	79-81	85-87	6
4	" " - 8 mins; " " " "	87-89	97-99	10
5	Peened, hand hammered.	86-88	98-100	12
6	" " "	89-91	104-108	17
7	Cloudburated - 1/2 meter drop; 30 mins. time.	79-81	85-87*	6
8	" - 1 meter drop; " "	76-78	86-88*	10
9	" - 2 meter drop; " "	84-86	93-95*	9
10	Elongated in Tensile Machine - 18% in 2 inches	80-82	96-98	16
11	" " " - 35% in 2 inches	80-82	97-99	17

* Specimen 7 - Pendulum Hardness After "Cloudbursting" = 337.
Specimen 8 - " " " " = 390.
Specimen 9 - " " " " = 378-472

sufficiently severe, can also develop the dendritic structure of the material. The hardness figures, together with the irregular appearance of the resulting macrograins abutting the surface of specimen 9 indicate that the two meter drop had produced a work-hardening beyond the capacity of the material for work hardening. Specimens 10 and 11 were machined to a standard

finish from 1¼ inch round bars extended 15 and 35 per cent respectively, in a 2-inch gage length in a 100,000-pound Riehle Testing Machine. Specimens 12, 13, 14, and 15 were not work-hardened; they are comparison specimens. No data is shown in the tables for specimen 12. This specimen had a surface such as is obtained from a drastic overpickling and it was included in the test along with the other specimens out of curiosity to see how it would resist corrosion. The comparison specimens 13, 14, and 15 had the regular No. 00 standard finish. Three other specimens, not shown in the tables, were given a 12-inch bastard flat file finish and exposed for 500 continuous hours to the salt-spray test to determine what influence the coarse finish as well as the accompanying cold work had upon its corrodibility.

CLEANING OF SPECIMENS.

Before exposure to the test, the specimens were cleaned with petrolic ether, then with alcohol, and placed in a desiccator to dry. The dried specimens were weighed to one-tenth of a milligram.

After exposure to the test, all soluble salts were removed from each specimen by holding it under a flowing stream of warm tap water. Appreciably large deposits of corrosion products were removed with a blunt edge tool. All other adherent corrosion products were loosened chemically, by immersion in cold concentrated nitric acid. Stain markings were taken off with a soft pencil eraser. The cleaned specimens were washed in warm water, immersed in alcohol and ether, cooled and dried in a desiccator, and weighed. After cleaning, the dried specimens were weighed and the process of cleaning and weighing repeated until a cleaning loss was established. The cleaning loss was then subtracted from the total loss in weight of the specimen and the remainder reported as the loss in weight caused by corrosion.

EXAMINATION OF TEST DATA.

(*a*) Salt-spray Test—The data contained in Table III shows that, with respect to surface appearance, the passivated and hand-hammered specimen 5 was the worst, having covered its conical and cylindrical surfaces with conspicuously large deposits of rust. A similar specimen, No. 6, which had been hand-hammered to a slightly higher value of hardness, however, showed very little vis-

Table III - Corrosion of Corrosion Resisting Steel, Grade 1, in 4% Sea Salt Spray

Original Data				Corrosion Losses After 2000-Hr. Exposure To 4% Salt Spray									
Specimen No.	Area Cm^2	Weight G.	Passivated*	Cleaning Loss Mg.	Corrosion Losses Mg.	Mg/Dm^2/Day	Mg/Cm^2/Month	Appearance of Specimen**					
								Cone		Cylinder		Pedestal	
								No.	Area	No.	Area	No.	Area
1	41.92	134.9704	No	0.3	3.5	0.10	0.03	3	1	2	1	1	5
2	42.00	136.3241	No	0.2	1.8	0.05	0.02	1	1	3	4	1	1
3	42.06	138.2744	Yes	0.2	1.4	0.04	0.01	0	0	4	1	0	0
4	42.03	137.1818	Yes	0.1	1.3	0.04	0.01	0	0	0	0	2	2
5	42.00	136.1492	Yes	0.1	1.7	0.05	0.01	1	7	1	4	1	4
6	42.07	138.4258	Yes	0.1	0.9	0.03	0.01	0	0	2	1	0	0
7	42.06	138.2564	Yes	0.0	0.6	0.02	0.01	0	0	0	0	0	0
8	42.06	138.2699	No	0.2	1.2	0.03	0.01	1	3	0	0	1	5
9	41.96	135.7409	Yes	0.1	0.6	0.02	0.01	1	3	0	0	0	0
10	41.94	135.2337	Yes	0.0	0.4	0.01	0.00	0	0	0	0	0	0
11	42.02	137.7924	No	0.0	0.5	0.01	0.00	0	0	0	0	0	0
13	42.07	138.5204	Yes	0.1	1.4	0.04	0.01	0	0	2	3	0	0
14	41.95	135.3028	No	0.2	0.6	0.02	0.01	0	0	0	0	1	5
15	42.02	137.0156	Yes	0.0	0.5	0.01	0.00	0	0	0	0	0	0

* Passivated in cold 50% conc. nitric acid for 1 hour.

** The numbers signify the number of rusted areas.

0 signifies no visible corrosion.

1 connotes a barely perceptible pin-point of discoloration.

2 " an area of rusting or staining approximately 1/64th of an inch in dia.

3 " " " " " " " " 1/32nd " " " " "

4 " " " " " " " " 1/16th " " " " "

5 " " " " " " " " 1/8th " " " " "

6 " " " " " " " " 1/4th " " " " "

7 " " " " " " " greater than 1/4th " " " " "

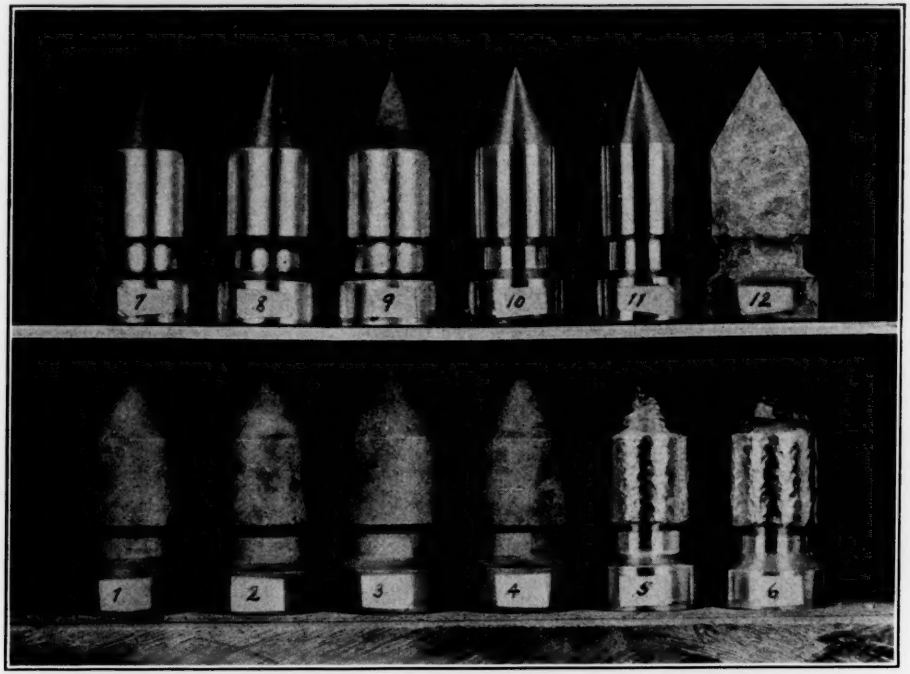


FIGURE 1.—CORROSION RESISTING STEEL, TITANIUM-BEARING, GRADE 1, AFTER 2000 HOURS IN 4 PER CENT SEA SALT SPRAY.

1–2–3–4.....Sandblasted.
5–6.....Peened.
7–8–9.....Cloudbursted.
10–11.....Elongated in tensile machine.
12.....Overpickled in 50:50 HCl.

ible corrosion. "Cloud-bursted" specimen 8 had also broken down in a few places. It, unlike its companion specimens 7 and 9, had not been passivated prior to exposure. The sandblasted specimens, though lightly stained in several places, looked very good considering the length of exposure. The specimens on which no corrosion was discernible even under a magnification of 7 diameters were 7, 10 and 15. In Figure 1 are shown twelve of the specimens after a 2000 hour exposure.

With respect to loss in weight, the sandblasted specimens suffered the most, this loss tending to decrease with increase of sandblasting time. Specimens 5 (hand-hammered), 8 ("cloud-bursted"), and 13 (standard comparison) likewise sustained comparatively heavy weight losses. The smallest loss in weight was shown by specimen 10 (elongated 15 per cent in tensile machine).

(*b*) Nitric Acid Test—The data presented in Table III indicates that the sandblasted specimens 2 and 3 were corroded most by the nitric acid, while the other four cold-worked specimens resisted

TABLE IV

Corrosion Losses
After
Exposure to 67% C.P. Nitric Acid for Three 48-Hour Periods

Specimen Number	(I) Cast CRS - Quenched from 2000°F in H_2O		(II) Same CRS as in Column (I) Plus Cold Working				
	Loss -Grams-	Penetration - Inches -	Cold Work	Hardened From Rockwell B	To	Loss -Grams-	Penetration - Inches -
2	.0324*	.00160	Sandblasted	80	86	.0575*	.00221
3	.0221	.00088	Sandblasted	83	90	.0545	.00205
5	.0620	.00249	Peened	78	92	.0285	.00105
6	.0268	.00121	Peened	81	97	.0235	.00090
10	.0284	.00117	Elong. 15%	81	97	.0271	.00101
11	.0284	.00117	Elong. 35%	81	98	.0251	.00094

*Total Loss after three 48-hour periods.
Dimension of specimens: 1/2"x 1/4"x 2".
Two specimens tested at one time.

acid attack more effectively; in fact, more effectively than did the same specimens when not cold-worked. The total corrosion after the third period, however, did in no instance exceed .004-inch penetration, which usually is the maximum allowable limit in many specifications.

CONCLUSIONS.

The following generalizations are substantiated by the results of these investigations. It is emphasized that these conclusions

1–2–3–4.....Sandblasted. 7–8–9.....Cloudbursted.
5–6.....Peened. 10–11.....Elongated in tensile machine.
12.....Overpickled in 50:50 HCl.

pertain only to corrosion resisting steel made and tested under the described method and conditions:

(1) Sandblasting: Due to the augmentation in effective area brought about by sandblasting, sandblasted surfaces have a noticeable tendency for increase in loss of weight by corrosion in either nitric acid or sea salt spray. A 2000-hour exposure to salt spray will most likely show some attack on the metal but practically no progressive corrosion. The appearance of the specimens, on the average, is fairly good and not as bad as loss in weight figures appear to indicate. The corrosion losses in nitric acid, though high, are not excessive and within most specification requirements.

(2) Peening: Since the corroding action varied from the faintest surface staining on specimen 6 to well beyond incipient rusting on specimen 5, it is stated that deforming the surface by hand-hammering may or may not lower the resistance of corrosion resisting steel to salt spray. At any rate, as verified by exposing three additional test specimens to the salt spray, the first 500 hours of exposure generally fail to develop any evidence of surface attack. The resistance to nitric acid is not lowered.

(3) Cloud-bursting: "Cloud-bursting" does not adversely affect corrosion quality in the salt spray test. Specimen 7, after 2000 hours of exposure, showed a corrosion rating equal to "A," representing an ideal condition and being practically perfect as regards to corrosion under the prescribed corrosion test; specimen 9 disclosed a condition almost equal to that of "A"; specimen 8 showed some attack on the metal.

(4) Cold Drawing: Corrosion resisting steel which has been permanently extended or cold drawn over its entire cross sectional area, suffers no discernible attack when subjected to the standard 4 per cent sea salt spray. Specimens 10 and 11 were entirely free from staining and rusting. They also showed a high resistance to chemical dissolution in nitric acid, higher than that shown by the same specimens not plastically deformed.

(5) Filing: Although no quantitative measurements were taken, visual inspection of three coarsely filed specimens revealed no trace of rust or stain after 500 hours of exposure.

(6) Passivation: Regarding passivation, it is stated that passivating treatments in cold 50 per cent concentrated nitric acid prior

to exposure tend to diminish the rusting and staining of cold-worked corrosion resisting steel in salt spray.

SUMMARY.

Cold work rarely lowers the resistance of corrosion resisting steel to corrosion in either nitric acid or salt spray. Provided they be clean—freshly filed, inadvertently bent, hammer bruised, and sandblasted materials—having the chemical composition shown in Table I, are therefore not likely to rust when exposed to the action of salt spray.

A METHOD FOR DETERMINING PROPER PITCH FOR THE INBOARD AND OUTBOARD PROPELLERS ON A FOUR SCREW SHIP.

By Lieutenant Commander J. M. Labberton, U.S.N.R., Member.*

While it has been known for many years that with identical propellers, the outboard screws of a four screw ship turn faster than the inboard screws, and that there is in consequence an unbalance on the power demand for the different shafts, it is only recently that the question of balancing the power demand on the four shafts has become interesting to marine designers. The desirability of doing this must be carefully weighed against the obvious advantage of having all propellers to the same diameter and pitch because of the expense of spares, but if after such consideration, a balance is still desired, the author deduces a method for modifying the pitch ratios.

Those who are familiar with Mr. Labberton's contributions to gear and propeller design will be interested to know that he has recently severed his connection with the Design Division of the Bureau of Engineering to assume the duties of Assistant Professor of Mechanical Engineering at New York University.

As a result of the variation in wake and thrust deduction between outboard and inboard propellers of quadruple screw ships, the outboard screws on a conventional hull, if identical to the inboard screws and powered by identical engines may operate at three to four per cent in excess of the R.P.M. of the inboard screws. A method of calculating the change in pitch necessary to have all propellers absorb the same power at the same R.P.M. thereby more closely meeting the specifications for identical engines is the subject of this paper.

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At the present time, we have no method of separating the total thrust deduction into the proper proportions, inboard and outboard. However, as regards wake fraction, it is a fairly easy matter when the facilities for self-propulsion of models are available. Methods of determining this wake fraction are outside the scope of this discussion.

As an example, let us consider a high speed merchant ship to be capable of 30 knots and requiring an effective horsepower of 88,000. Assume that a preliminary self-propelled test has been made on the model and that calculations therefrom indicate that

Wake Fraction inboard = .165
Wake Fraction outboard = .075
Thrust deduction = .18

The propulsion machinery is designed to operate at 192.5 R.P.M. and it is desired to have both inboard and outboard shafts turning at equal revolutions and absorbing the same power.

In order to compare the effects of equal pitch on all propellers and corrected pitch according to position, a propeller will first be designed as a compromise for all positions and the performance worked out for each position. The same will then be done for corrected pitches.

As a basis of design, the data in Admiral D. W. Taylor's "The Speed and Power of Ships" will be used. Refer to Figure 208 in this book. Since cavitation will certainly be a problem, the blades will be made the maximum width practicable, that is with a Mean Width Ratio = .50.

Considering the strength necessary, the Blade thickness fraction is made .06. With this combination of M.W.R. and B.T.F. a correction factor of 1.017 is necessary for Delta and a correction factor of .975 is necessary for efficiency on Figure 208.

$$\text{Knots} = V = 30$$

$$\text{E.H.P./Propeller} = \frac{88{,}000}{4} = 22{,}000$$

$$\text{Thrust Deduction} = t = .18$$

$$\text{Wake Fraction} = W = \frac{.165 + .075}{2} = .12$$

$$\text{R.P.M.} = N = 192.5$$

$$\text{Speed of Advance} = V_a = V\,(1 - W)$$

$$V_a = 30\,(1 - .12) = 26.4$$

$$V_a^{5/2} = 3581$$

$$\text{Useful Horsepower} = U = \text{E.H.P.}\ \frac{1 - W}{1 - t}$$

$$U = 22{,}000 \times \frac{1 - .12}{1 - .18} = 23{,}610$$

$$U^{1/2} = 153.7$$

$$BU_3 = \frac{NU^{1/2}}{V_a^{5/2}} = \frac{192.5 \times 153.7}{3581} = 8.26$$

Selecting the pitch ratio for maximum efficiency from Figure 208, values read as follows:

$$\text{Pitch Ratio} = a = 1$$

$$\text{Delta} = 129 \times 1.017 = 131 \text{ corrected.}$$

$$\text{Efficiency} = .705 \times .975 = .69 \text{ corrected.}$$

Therefore

$$\text{Propulsive coefficient} = \text{Efficiency}\ \frac{1 - t}{1 - W}$$

$$\text{or P. C.} = .69\ \frac{.82}{.88} = .64$$

$$\text{Shaft Horsepower required for each shaft} = \frac{22{,}000}{.64} = 34{,}375$$

The diameter of the propeller, D is determined as follows:

$$D = \frac{\text{Delta} \times \text{Va}}{N}$$

$$= \frac{131 \times 26.4}{192.5} = 17.965$$

$$D = 18 \text{ feet.}$$

Pitch = 18 feet.

The performance at each position will now be worked out.

OUTBOARD.

$V = 30$

$E.H.P. = 22{,}000$

$t = .18$

$W = .075$

$V_a = 30\ (1 - .075) = 27.75$

$V_a^{5/2} = 4052$

$$U = 22{,}000\ \frac{.925}{.82} = 24{,}817$$

$U^{1/2} = 157.5$

$$BU_3 = \frac{NU^{1/2}}{V_a^{5/2}} = \frac{192.5 \times 157.5}{4052} = 7.46$$

Referring to Figure 208, it is seen that for a = 1

Efficiency = $.975 \times .715 = .70$ corrected

$$P.C. = .70\ \frac{.82}{.88} = .65$$

$$S.H.P. = \frac{22{,}000}{.65} = 33{,}846$$

Thus less than the 34,375 Shaft Horsepower is used at 192.5 R.P.M. and the shaft will tend to speed up unless throttled.

INBOARD.

$V = 30$

$E.H.P. = 22{,}000$

$t = .18$

$W = .165$

$V_a = 30\ (1 - .165) = 25.05$

$V_a^{5/2} = 3140$

$$U = 22{,}000 \frac{.835}{.82} = 22{,}402$$

$$U^{1/2} = 149.7$$

$$BU_3 = \frac{NU^{1/2}}{V_a^{5/2}} = \frac{192.5 \times 149.7}{3140} = 9.22$$

Referring to Figure 208, it is seen that for a = 1

Efficiency = $.975 \times .695 = .68$ corrected

$$\text{P. C.} = .68 \frac{.82}{.88} = .63$$

$$\text{S. H. P.} = \frac{22{,}000}{.63} = 34{,}921$$

Thus more than the 34,375 Shaft Horsepower is required at 192.5 R.P.M. and this shaft will tend to run at lower speed.

If the speed-torque characteristics of the propulsion turbines were known, the actual revolutions and horsepower outputs of the inboard and outboard turbines could be determined by calculation.

Propellers will now be designed having different pitches inboard and outboard and so designed as to have inboard and outboard shafts revolving at the same speed and absorbing the same power.

In order to do this, a preliminary design must be made in order to obtain an indication as to efficiency.

OUTBOARD

$$V = 30$$

$$\text{E. H. P./ Propeller} = \frac{88{,}000}{4} = 22{,}000$$

$$t = .18$$

$$W = .075$$

$$N = 192.5$$

$$V = 30\ (1 - .075) = 27.75$$

$$V_a^{5/2} = 4052$$

$$U = 22{,}000 \frac{.925}{.82} = 24{,}817$$

$U^{1/2} = 157.5$

$BU_3 = \frac{NU^{1/2}}{V_a^{5/2}} = \frac{192.5 \times 27.75}{4052} = 7.48$

Refer to Taylor's Figure 208

a = 1.025

ep = $.975 \times .717 = .70$

Delta = $1.017 \times 123.8 = 126$

$d = \frac{\text{Delta} \times V_a}{N} = \frac{126 \times 27.75}{192.5} = 18.16$

$\text{P. C.} = .70 \frac{.82}{.925} = .70 \times .886 = .62$

This indicates an S. H. P. of $\frac{22,000}{.62} = 35,484$

INBOARD.

V = 30

E. H. P./Propeller = 22,000

t = .18

W = .165

N = 192.5

$V_a = 30 (1 - .165) = 25.05$

$V_a^{5/2} = 3140$

$U = 22,000 \frac{.835}{.82} = 22,402$

$U^{1/2} = 149.7$

$BU_3 = \frac{NU^{1/2}}{V_a^{5/2}} = \frac{192.5 \times 149.7}{3140} = 9.18$

Refer to Taylor's Figure 208

a = .955

ep = $.975 \times .697 = .68$

Delta = $1.017 \times 136.8 = 139$

$$d = \frac{\text{Delta} \times V_a}{N} = \frac{139 \times 25.05}{192.5} = 18.09$$

$$\text{P.C.} = .68 \frac{.82}{.835} = .67$$

$$\text{This indicates a S.H.P. of } \frac{22{,}000}{.67} = 32{,}836$$

$$\text{Take average } P = \frac{35{,}484 + 32{,}836}{2} = 34{,}160$$

$$\text{Take diameter} = \frac{18.16 + 18.09}{2} = 18.125$$

Making use of this data, a set of propellers will now be worked through by the " Power " method.

Refer to Taylor's Figure 204.

OUTBOARD.

$V = 30$

$P = 34{,}160$

$t = .18$

$W = .075$

$N = 192.5$

$V_a = 30\ (1 - .075) = 27.75$

$V_a^{5/2} = 4052$

$P^{1/2} = 184.8$

$$Bp_3 = \frac{NP^{1/2}}{V_a^{5/2}} = \frac{192.5 \times 184.8}{4052} = 8.78$$

$d = 18.125$ (from above)

$$d = \frac{\text{Delta} \times V_a}{N}$$

$$18.125 = \frac{\text{Delta} \times 27.75}{192.5}$$

Delta $= 125.73$ (for M.W.R. $= 50$, B.T.F. $= .06$)

Delta $= \frac{125.73}{1.017} = 123.6$ (for M.W.R. = .25, B.T.F. = .05)

Refer to Taylor's Figure 204.

a = 1.03

ep = .975 × .718 = .70

Pitch = 1.03 × 18.125 = 18.67 feet

P.C. $= .70 \frac{.82}{.925} = .62$

E.H.P. = 34,160 × .62 = 21,179

INBOARD.

V = 30

P = 34,160

t = .18

W = .165

N = 192.5

$V_a = 30 (1 - .165) = 25.05$

$V_a^{5/2} = 3140$

$P^{1/2} = 184.8$

$Bp_3 = \frac{NP^{1/2}}{V_a^{5/2}} = \frac{192.5 \times 184.8}{3140} = 11.33$

d = 18.125 (from above)

$d = \frac{\text{Delta} \times V_a}{N} = \frac{\text{Delta} \times 25.05}{192.5} = 18.125$

Delta = 139.28 (for M.W.R. = .50, B.T.F. = .06)

Delta $= \frac{139.28}{1.017} = 137$ (for M.W.R. = .25, B.T.F. = .05)

Refer to Taylor's Figure 204.

a = .973

ep = .975 × .692 = .675

Pitch = 17.636 feet

$$\text{P.C.} = .675\,\frac{.82}{.835} = .663$$

$$\text{E. H. P.} = 34{,}160 \times .663 = 22{,}648$$

Total E. H. P. delivered to the hull is:

$$2 \times 21{,}179 + 2 \times 22{,}648 = 87{,}654$$

$$\frac{87{,}654}{88{,}000} = 99.607 \text{ per cent}$$

Less than 4/10 of one per cent off.

Total S. H. P. required $= 4 \times 34{,}160 = 136{,}640$

The percentage difference in pitches of the propellers is

$$\frac{18.67}{17.636} = 1.06 \text{ or } 6 \text{ per cent}$$

which is quite different from the percentage difference in the speeds of advance which is

$$\frac{27.75}{25.05} = 1.11 \text{ or } 11 \text{ per cent}$$

Tests of self-propelled models fitted with propellers with differences in pitch calculated in accordance with the above method give very nearly the same power and R.P.M. for all shafts. It should be noted that the percentage difference in pitch is the important factor rather than the actual pitch.

THE INCREASE IN FRICTIONAL RESISTANCE DUE TO THE ACTION OF WATER ON BOTTOM PAINT.

By E. A. Stevens, Jr., Member.*

In this brief discussion, Mr. Stevens points out the rather startling effect of the aging of bottom paints in water on the hull frictional resistance factor of ships. While the data he offers, and the conclusions he reaches are open to attack not only because of the rarity of opportunity to secure such data but because of the confusing factors which affect the results obtained on any full scale ship trial at sea, a brief consideration will convince the most skeptical that some increase from simple roughening of the bottom is to be expected, and Mr. Stevens presents enough points to indicate the trend of the curve. While the curve is presumed to represent naval rather than merchant marine employment, some consideration of the type of employment will indicate how it can be expected to vary.

Water acts on paints used on ship's bottoms causing it to roughen, thereby materially increasing the frictional resistance even though there be no marine growth. This increase is quite rapid at first but slows up as the time out of drydock lengthens, providing no marine growth forms. Figure 1 shows the percentage increase as obtained from analyses of the trials of several Navy vessels. The abscissas are the number of days the ship was in the water between drydocking and the trial, the ordinates representing the percentage increase in frictional resistance over that of a freshly painted bottom.

The data for this curve were obtained as follows: Trial results of a ship with a freshly painted bottom were analyzed by the "Dyson" method at various speeds for the power augment factor K. Using these values of K for analyzing the results of another trial of the same ship, after she had been in the water for some

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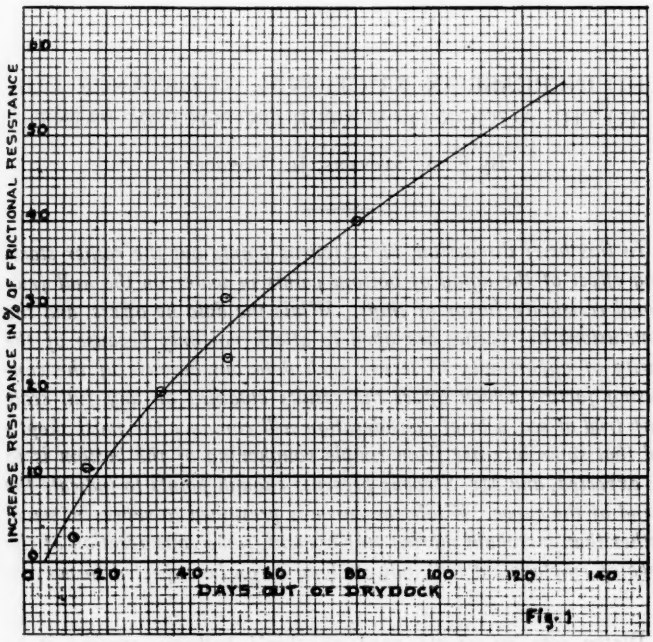

Fig. 1

time, the effective horsepower for various speeds were obtained. The difference between the effective horsepower obtained from the latter trial and that obtained from the tank test is the increase due to the action of the water on the bottom paint. Knowing the power required to overcome the frictional resistance, the percentage increase is readily obtained.

All of the ships used in obtaining the data for this curve were Navy vessels, which are, as a rule, not under way as much of the time as those engaged in the merchant service, therefore, it is quite probable that the increase frictional resistance of merchant vessels after they have been in the water twenty (20) to thirty (30) days is somewhat less than that shown on Figure 1. An analysis of a year's operation of the T. S. S. *President Harding* in the North Atlantic trade showed an average increase effective horsepower of 1350 over that given by the tank test. As this ship was drydocked

every six months the average increase due to additional friction may be taken as equivalent to eighty (80) to ninety (90) days in the water. Figure 1 shows this to be 45 per cent of the frictional resistance, which in this case, would amount to an additional effective horsepower of 2230, which is considerably greater than the total increase due to all causes. Thus, it can be readily seen that, when a ship is under way a large part of the time the action of the water on the bottom paint is not as detrimental as is the case of a ship lying idle for most of the time.

Most bottom paints contain poisonous salts which are dissolvable in water. These salts kill any marine growth that may attach themselves to the ship's bottom and when in motion the dead growth and the outer layer of paint are washed away, thus exposing a new layer. If the ship is idle for any length of time, what is left of the outer layer of paint and the dead marine growth adheres to the bottom, thereby causing a rougher surface.

The increase frictional resistance of merchant ships which are in continuous operation in cool water, such as the North Atlantic, probably never exceeds 20 per cent of the original frictional resistance if they are drydocked every six to seven months and their bottoms kept in good condition. The average increase under such conditions is probably not over 15 to 17 per cent. In tropical waters, however, fouling will occur in a short time on an idle ship, even if the best of anti-fouling paints are used, which may cause an increase of 25 to 30 per cent or more.

The friction resistance often increases as a ship ages. This is caused by roughening of the ship's bottom due to pitting of the plates, or by applying new paint over old which has become rough. There are some cases which indicate that the increase due to this cause is as much as 20 per cent of the original frictional resistance. However, the available data is not as satisfactory as may be desired and the above percentage should be considered only as a rough approximation. If a ship's bottom has been kept in good condition, the increase due to her age will not be great, but should the bottom be allowed to become rough, a considerable increase resistance is to be expected.

With the exception of tankers and some bulk carriers, it is very seldom possible to conduct loaded trials of a merchant ship until

she has been in the water for twelve (12) to twenty (20) days. In such cases allowances should be made for the increase frictional resistance due to this period in the water and as the ship is idle most of the time, being at a dock loading, the percentage increase can be taken directly from Figure 1.

The following example will illustrate how the allowances for increase frictional resistance is made:

EXAMPLE.

A ship has been in the water twenty (20) days when the runs on the measured mile were made. From Figure 1 the percentage increase is 12 per cent. The model having been tested in the tank the frictional and total effective horsepower are obtainable.

I	Speed	9	10	11	12
II	Fric. E.H.P. from Tank Test	500	680	890	1140
III	Fric. E.H.P. $\times$ .12 = II $\times$ .12	60	82	107	137
IV	Total E.H.P. Tank Test	660	900	1205	1630
V	Total E.H.P. on Trial = III + IV	720	982	1312	1767

It should be noticed that the curve on Figure 1 shows no increase in frictional resistance until the ship has been in the water five (5) days. As it is very seldom possible to run trials on a measured mile within three (3) to five (5) days after the ship has been undocked, five (5) days were taken as the starting point.

NOTES.

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Metallurgy and the Aero-Engine.
—The Engineer, London, England, September 10, 1937.

Fuels of Today and Tomorrow.
—Combustion, New York, N. Y., August, 1937.

Miscellanea of Interest to Naval Engineers.

Water Circulation in Boilers.
—The Engineer, London, England, October 8, 1937.

Inspecting Without Doing Damage.
—The General Electric Review, Schenectady, N. Y., October, 1937.

Almost Elemental.
—The Industrial Bulletin, Arthur D. Little, Inc., Cambridge, Mass., September, 1937.

Deaerated Feedwater.
—Power, New York, N. Y., September, 1937.

Costs for New Destroyers.
—The Log, San Francisco, Cal., September, 1937.

Noise.
—Engineering, London, England, September 17, 1937.

Some Comments on Micrurgy.
—The General Electric Review, Schenectady, N. Y., October, 1937.

The New Motor Torpedo Boats.
—The Shipbuilder and Marine Engine Builder, London, England, October, 1937.

Metals We Eat.
—Mining and Metallurgy, New York, N. Y., October, 1937.

U. S. NAVY PARTICIPATION IN DIESEL ENGINE DEVELOPMENT.

At the Tenth National Oil and Power Meeting of the American Society of Mechanical Engineers, held at State College, Pa., August 18-21, 1937, Mr. E. C. Magdeburger of the Bureau of Engineering, Navy Department, presented the paper of which the following is an abstract, drawn in large part from Mechanical Engineering, September, 1937. It contains a valuable summary of foreign and U. S. Diesel development at the present time, and will be found useful for reference purposes.

Efficiency is a divine attribute and the foundation of every virtue in an engineer. The engine, now universally designated by the name of the originator of the idea to make the heat of compression ignite the fuel charge, is the result of a systematic search on his part for a more efficient way of converting the heat within the fuel into mechanical energy. In the forty years of its successful development no one worthy of the name of engineer challenged this principal claim of the Diesel engine that it represents the most efficient means of producing mechanical energy by combustion, i.e., more power can be produced out of a given weight of fuel oil in a Diesel engine than in any other way.

Other important advantages possessed by the Diesel engine from a naval engineer's point of view include

1. Ability to start quickly and develop full load in less than one minute, if necessary.
2. Smokeless exhaust under all normal operating conditions, a great advantage for submarines and aircraft carriers.
3. Economy in space required within the ship, particularly if the bulky boiler uptakes are taken into consideration.
4. Weight per shaft horsepower comparing favorably with the best of the steam plants of equal reliability.
5. Reliability equaling the proverbially reliable though far less economical types of steam plant.
6. Availability far surpassing that of any steam plant as demonstrated by modern streamlined trains.
7. Vulnerability far less than any steam plant, which at times requires high-pressure steam piping of excessive length, the latter proved to be a source of great danger in naval engagements of the World War.
8. Adaptability to the propelling-plant requirements peculiar to a naval vessel.

On the other hand, anyone even casually coming in contact with the combatant ships of the Navy is bound to be impressed by the tremendous efficiency of their propelling plants in weight and space required. Naval propelling plants, whatever their type, represent the highest development of the art consistent with the desired degree of reliability and life expectancy.

GERMANY LEADS IN DEVELOPING NAVAL DIESEL ENGINES.

How far then did the navies of the world appreciate the Diesel engine and how much did the United States Navy contribute to its development, particularly within the last few years? This question is believed to be of interest to all engineers and the author will therefore attempt to throw some light on the subject even though a veil of secrecy must, for obvious reasons, hide the details of the picture as far as it concerns the recent Diesel-engine applications in the United States Navy.

When building of naval vessels was resumed in Germany the development of the modern high-speed, lightweight types of Diesel engine was not sufficiently advanced to permit their consideration for propelling plants of warships. Not until after the three cruisers, *Königsberg, Karlsruhe* and *Köln*, were well under way did test results with an M.A.N. four-cycle, two-cylinder experimental unit, developing 100 B.H.P. per cylinder at 1000 R.P.M. and weighing 11 pounds per H.P., give sufficient promise to permit the installation of two 10-cylinder cruising units geared to the propeller shafts. A hydraulic clutch of the Foettinger type, interposed between the engine and its reduction gear, Figure 1, permits the Diesel engine to be easily and

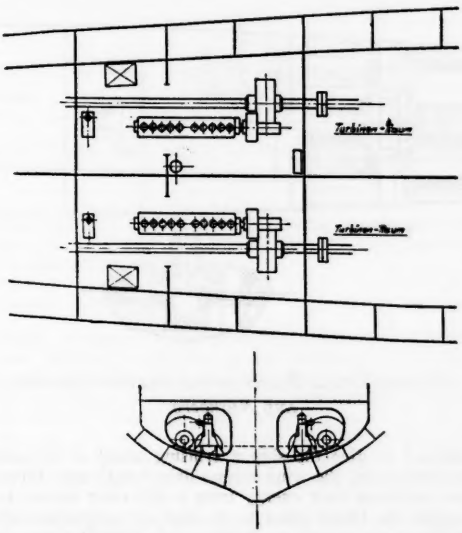

FIG. 1. CRUISING DIESEL PLANT OF THE GERMAN CRUISERS, *Königsberg, Karlsruhe,* AND *Köln.*

reliably disconnected when steam turbines are to be operated. These Diesel engines were thus installed for cruising only and must be carried as dead weight when steam turbines are in operation. To avoid excessive upkeep the rating of the Diesel engines had to be reduced later to 900 B.H.P. at 900 R.P.M., which gave a ship speed of about 10 knots and a cruising radius of about 18,000 miles with all bunkers filled with Diesel oil.

These Diesel engines appear much too small to be of real value to the ship, since on long cruises foulness of the bottom reduces the ship's speed materially. Besides in wartime even on long cruises, a substantial part of the boiler plant must be kept under steam and the turbines warmed up if a reasonably quick getaway is to be assured. The steam plant of these cruisers is designed for a maximum output of 60,000 S.H.P.

German Cruisers *Leipzig* and *Nürnberg.*

Meanwhile, the development by M.A.N. of a four-cylinder two-cycle double-acting experimental unit of 250 B.H.P. per cylinder at 800 R.P.M. and weighing only 6.6 pounds per H.P., without blower, permitted approaching the problem of including Diesel engines into a naval propelling plant from a different point of view. Therefore, in the cruiser *Leipzig,* although the 60,000 S.H.P. steam-turbine twin-screw plant was retained, no cruising turbines were provided and a 12,000-S.H.P. Diesel propelling unit was added to operate a separate propeller in the center, Figure 2. Four Diesel engines were connected to a common reduction gear by hydraulic clutches

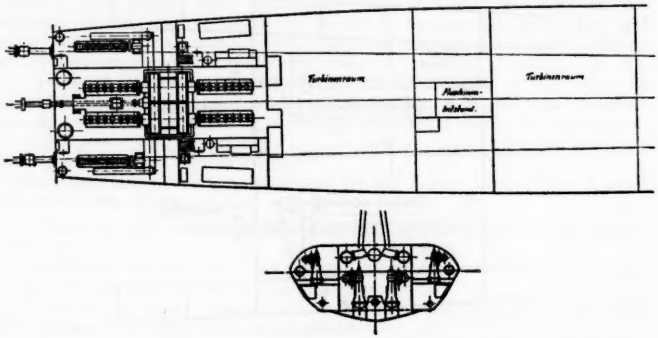

FIG. 2. CRUISING DIESEL PLANT OF THE GERMAN CRUISERS, *Leipzig* AND *Nürnberg.*

and were designed to give the ship a cruising speed of 18 knots with the turbines disconnected and the wing screws driven with zero thrust by geared electric motors receiving their energy from a generator geared to the center shaft. To permit the Diesel plant to be used for propulsion when turbines were driving the wing screws a variable-pitch propeller was fitted to the center shaft. In addition to the four seven-cylinder propelling Diesel engines two auxiliary seven-cylinder Diesel units of smaller cylinder dimensions but also of two-cycle double-acting type were installed. These drive the scavenging blowers and circulating-water and lubricating-oil pumps as well as a starting air compressor. Later a similar plant was installed in the cruiser *Nürnberg,* which is substantially a sister ship of the *Leipzig.*

German Gunnery Training Ship *Bremse.*

While in these two ships a composite propelling plant was employed, the next German warship to be built, the gunnery training ship *Bremse,* became the first completely Diesel-driven warship, excepting submarines of course. The principal reason for the adoption of a Diesel propelling plant was the fact that the exceedingly light construction of this vessel permitted no protection for steam piping from the boiler to the turbine, and an unprotected pipe carrying high-pressure, high-temperature steam is a source of real

danger in a war vessel. A total of 26,000 S.H.P. in two four-engine geared units, using eight-cylinder engines of the same cylinder dimensions as those of the main engines in the *Leipzig*, was installed, Figure 3.

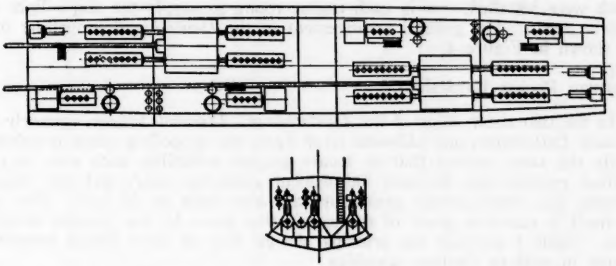

FIG. 3. DIESEL PROPELLING PLANT OF THE GERMAN GUNNERY TRAINING SHIP *Bremse*.

For the four auxiliary engines driving scavenging blowers and pumps the same cylinder size was used but these units had only four cylinders and operated at lower speed.

Pocket Battleship *Deutschland*.

The successful progress of the preceding program of Diesel-engine construction gave sufficient confidence in the feasibility of a still larger and more important propelling plant, so that a twin-screw 54,000-S.H.P. Diesel plant for the so-called pocket battleship formerly known as *Ersatz Preussen* and later christened *Deutschland* was decided upon. The plant consists as in the *Bremse* of two four-engine geared units using nine-cylinder engines of almost double the horsepower output per cylinder, a cylinder size which was meanwhile thoroughly tested out experimentally. The four auxiliary

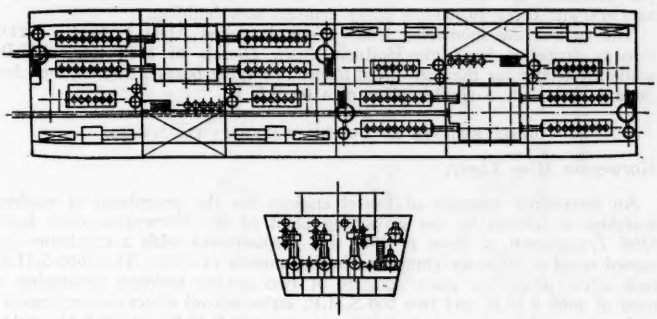

FIG. 4. DIESEL PROPELLING PLANT OF THE GERMAN POCKET BATTLESHIP *Deutschland*.

engines driving the scavenging blowers, as well as circulating-water and lubricating-oil pumps, consist of five-cylinder units of the same cylinder dimensions but with lower maximum speed—425 R.P.M. In addition, eight four-cycle Linke-Hoffman-Busch Diesel-driven generators of 250 Kw. each were installed, two in each engine room, to supply the ship's light and power needs. The general arrangement of the *Deutschland* propelling plant is shown in Figure 4.

German Pocket Battleships.

In the two sister ships of the *Deutschland, Admirial Scheer,* formerly the *Ersatz Lothringen,* and *Admiral Graf Spee,* the propelling plant is substantially the same, except that to secure greater reliability each pair of propelling engines has its own independent reduction gear, and the engines driving the ships' service generators are also built by M.A.N. This fact in itself is sufficient proof of the satisfaction given by the original installation. Table 1 presents the principal design data of these Diesel propelling plants in modern German warships.

Many interesting details in the design of the Diesel engines are used in the propelling plants of German warships. Pertinent numerical design data will be found in Table 2. However, only such other details will be briefly enumerated as may be of interest from an operator's point of view. All engines are directly reversible and all four engines, geared to one propeller shaft, can be maneuvered as a group by one lever or individually if necessary. Only upper cylinders are equipped with air starting valves. One fuel valve is used in the upper cylinder and two in the lower. Rotary valves control the exhaust ports. All pistons of double-acting engines have forged-steel heads and are oil cooled, while salt water is used for cylinder jackets and exhaust header. Engine framing is of welded-steel construction. The cylinder consists of a cast-iron central part carrying the scavenging and exhaust ports for both ends and two similar thimble-shaped steel end pieces with separate thin welded-steel jackets. Airless injection of fuel is used consistently.

The two 26,000-ton German battleships *Scharnhorst,* launched October 3, 1936, and *Gneisenau,* launched December 8, 1936, are reported to have a Diesel cruising plant of unknown characteristics, while no information is yet available for the two 35,000-ton battleships, two 19,250-ton aircraft carriers, and three 10,000-ton heavy cruisers now building.

The twin-screw submarine tender *Saar,* launched April 5, 1934, of 2710 tons is propelled by Linke-Hoffman-Busch Diesels of 3700 total S.H.P., while the tender for the high-speed torpedo-carrying boats *Tsingtau,* launched June 6, 1934, has a similar plant of 4100 S.H.P.

DIESEL ENGINES IN MODERN WARSHIPS ELSEWHERE.

Norwegian Mine Layer.

An interesting example of Diesel engines for the propulsion of modern warships is offered by the propelling plant of the Norwegian mine layer *Olaf Tryggvason,* a vessel of 1924 tons displacement with a maximum designed speed of 20 knots, completed in the summer of 1934. The 6000-S.H.P. twin-screw propelling plant consists of two geared turbines developing a total of 4600 S.H.P. and two 700-S.H.P. series-wound direct-current motors each connected to a generator driven by an 850-B.H.P., 530-R.P.M. eight-cylinder, four-cycle, Sulzer Diesel engine. Fourteen knots can be obtained on Diesel engines alone and fuel supply for a cruising radius of 3000 miles

TABLE 1 DESIGN DATA FOR DIESEL ENGINES IN MODERN GERMAN WARSHIPS

Name of vessel	*Königsberg*	*Karlsruhe*	*Köln*	*Leipzig*	*Nürnberg*	*Bremse*	*Deutschland*	*Admiral Scheer*	*Admiral Graf Spee*
Date of launching	3-26-27	8-20-27	5-23-28	10-18-29	12-8-34	1-24-31	5-19-31	4-1-32	6-30-34
Date of completion	1929	1929	1929	1931	Nov. 1935		1933	1934	1936
Total propelling power of Diesel engines, shp	1,600			12,000		26,000		54,000	
Maximum speed on Diesel engines, knots	10			18		27		26	
Number of engines	2			4		4		8	
Power of each engine, bhp	900			3,100		3,550		7,100	
Type of engine	Single-acting four-cycle M.A.N., 10 × 26/33[a]			Double-acting two-cycle M.A.N., 7 × 30/44[b]		Double-acting two-cycle M.A.N., 8 × 30/44[c]	Double-acting two-cycle M.A.N., 9 × 42/58[d]		
Number of auxiliary engines driving scavenging blowers and pumps				2		4		4	
Type				Double-acting, two-cycle M.A.N., 7 × 23/34		Double-acting, two-cycle M.A.N., 4 × 30/44		Double-acting, two-cycle M.A.N., 5 × 42/58	
Bore, in.				9.056		11.811		16.536	
Stroke, in.				13.386		17.323		22.835	
Speed, rpm				750		530		425	

[a] For details, see Table 2, line 1; [b] Table 2, line 2; [c] Table 2, line 3; [d] Table 2, line 4.

is provided. A maximum speed of 21.7 knots was attained on trials with both steam turbines and Diesel engines in operation. Astern turbines are provided and, therefore, maneuvering can be done with motors and turbines to permit either plant to operate independently. From the scant description of the plant available the following features of operation may be inferred: Exhaust gases of the Diesel engines pass through exhaust-gas boilers, which also serve as mufflers, and generate sufficient steam not only to keep the blading of the steam turbines "cool" but also at higher speeds undoubtedly permit the latter to supply power to the screws, thus improving the fuel economy and cruising radius of the Diesel plant and doing away with the necessity of keeping the regular steam boilers under pressure which appears to be worth-while. This plant follows the lines of that on the British 6790-ton 27¾-knot mine layer *Adventure,* completed in 1926, which has in addition to its 40,000-S.H.P. geared steam-turbine plant a Diesel-electric plant of unknown capacity for cruising purposes.

Finnish Gunboats.

Two twin-screw Finnish 4000-ton, 16-knot gunboats, *Vainamoinen,* completed late in 1932, and *Ilmarinen,* completed early in 1933, are also propelled by Diesel engines. Their plants are of 5000 S.H.P., use Diesel-electric drive, and consist of two generators driven by high-speed W-type four-cycle Germania Diesel engines, supercharged on the Büchi system and two motors, one on each shaft.

French Submarine Tender.

The French twin-screw submarine tender *Jules Verne* of 5747 tons, launched February 3, 1931, is propelled by two eight-cylinder two-cycle Sulzer Diesel engines developing a total of 7000 S.H.P. These engines are coupled directly to the propeller shafts. They are directly reversible, of the air-injection type, and at the time of their shop tests in 1925 created a sensation by developing 6000 B.H.P. at 280 R.P.M. in eight cylinders of 650-millimeter bore and 680-millimeter stroke.

Japanese Submarine Tender and Mine Layers.

The Japanese 10,000-ton submarine tender *Taigei,* launched November 16, 1933, is propelled by four sets of Diesel engines totaling 13,000 S.H.P. and developing 20 knots. The mine layer *Itukusima,* launched May 22, 1929, is a triple-screw vessel of 1970 tons displacement propelled by three Diesel engines of 1000 S.H.P. each. Another mine layer, the *Nasami,* which was launched on March 26, 1934, is of 443 tons and is equipped with two sets of Diesel engines totaling 2300 S.H.P. and develops a speed of 19 knots.

Polish Mine Layer.

The *Gryf* was built in France and was launched November 29, 1936. It is of 2227 tons and its two sets of Sulzer Diesel engines totaling 6000 S.H.P. give it a speed of 20 knots.

Swedish Icebreaker.

Sweden boasts the honor of having the first large icebreaker equipped with Diesel engines as a naval auxiliary. The *Ymer,* launched in 1933, is of 3465 tons displacement and develops 18 knots on its Atlas Diesel engines totaling 9000 S.H.P.

6000-B.H.P. Experimental Diesel Engines.

Agitation caused by the Diesel-engine propelling plants for the German pocket battleships resulted in the French Navy ordering from two different sources, Sulzer Bros. and M.A.N., a three-cylinder 6000-B.H.P. experimental unit with the reported purpose of having four 12-cylinder 25,000-B.H.P. units of this type for the direct drive of a large French warship. Eventually in 1931 both of these experimental units were completed and tested and some test data were published. The French Navy Department is believed to have obtained a license to build the M.A.N. engine of this size but no further progress of the project has so far been reported. This M.A.N. type 2000-H.P. cylinder size follows the essentials of the design that was used in the *Deutschland*.

Necessity is said to be the mother of invention, and the father of courage, I would like to add: Forced by the necessity of building new submarine tonnage of considerably increased speed, which required 2 H.P. to be developed on the weight allowed for 1 prior to that, Admiral S. M. Robinson, chief of the Bureau of Engineering, had the foresight and courage to take advantage of the then prevailing "depression" with resultant abundance of available engineering talent and great anxiety of all engine builders to keep their engineering organization intact and occupied.

HIGH-SPEED DIESEL-ENGINE PROGRAM OF U. S. NAVY.

In the beginning of March, 1932, a circular letter was therefore addressed to all Diesel-engine and several gasoline-engine builders inviting them to submit proposals for a unit consisting of a high-speed Diesel engine and a generator. The weight of such a unit was restricited to 27½ pounds per B.H.P. of the engine but no limitations as to type of cycle, number and disposition of cylinders, speed, and mean effective pressures were imposed.

By December, 1932, five companies had signed a contract with the Navy to deliver either a complete unit of desired size or a part of one capable of expansion into sizes desired by the Bureau. All of these units have passed their acceptance tests but of course with varying degrees of success as would be expected.

Frank B. Stearns of Cleveland, who prospered by developing the Knight sleeve-valve engine for the automobiles he had been building up to 1925, was interested in the development of a high-speed Diesel engine as a hobby and was the first to respond with a proposal to build a so-called "diamond type" engine of which he had a small-bore unit then running. Such unit is composed by superimposing two V-type engines so that the upper cylinders have a common combustion space with the lower. The unit contracted for had two diamonds or eight pistons of $5\frac{1}{4} \times 8\frac{1}{2}$ inches with a maximum rating of 321 B.H.P. at 1300 R.P.M. Details of this engine are given in Table 2, line 7.

After many difficulties the engine was accepted by the Bureau and was set up at the Engineering Experiment Station, Annapolis, Md., for further testing. However, a weakness in design developed and the engine wrecked itself, thereby concluding the development of this type. Although success had been denied Mr. Stearns, probably because he tackled too many problems without assembling a sufficiently qualified personnel to solve them, nevertheless he deserves credit for having served as a "catalyst," initiating events that led to success by others.

The Winton Engine Corporation undertook to build a 12-cylinder V-type two-cycle engine developing 950 B.H.P. at 720 R.P.M. and employing

TABLE 2 DESIGN DATA FOR MODERN NAVAL DIESEL ENGINES

Line no.	Type of engine	Cycle	Cylinders	Bhp Total	Bhp One cylinder	Bore, in.	Stroke, in.	Speed, rpm	Displacement, cu in.	Piston speed, fpm	Brake mep, lb per sq in.	Specific volume, cu. in. per bhp	Remarks
1	M.A.N. 10 × 26/33	4	10	900	90	10.236	12.992	900	10,660	1940	74.2	11.84	German Navy K-class cruisers
2	M.A.N. 7 × 30/44	2DA	7	3,100	443	11.811	17.323	600	27,580	1,732	74.2	8.8	*Leipzig*
3	M.A.N. 8 × 30/44	2DA	8	3,550	443	11.811	17.323	600	31,520	1,732	74.2	8.8	*Bremse*
4	M.A.N. 9 × 42/58	2DA	9	7,100	789	16.535	22.835	450	83,950	1,713	74.4	11.8	*Deutschland*
5	M.A.N. 3 × 64/96	2DA	3	6,000	2,000	25.197	37.795	263	107,500	1,655	84.0	17.9	French Navy experimental
6	Sulzer 3 × 68/94	2DA	3	6,400	2,133	26.772	37.009	267	118,600	1,646	80.0	18.54	
7	Stearns	2◇	4	321	80.25	5 1/4	8 1/2	1,300	1,470	1,840	66.6	4.58	
8	Winton	2V	12	950	79.17	8	10	720	6,040	1,200	86.6	6.36	
9	Sun Ship	2 O.P.	6	510	85.0	6 1/2	9 3/4	650	3,890	1,050	80.0	7.62	U. S. Navy experimental
10	Continental	2 Rad.	10	480	48.0	6 1/2	7 1/2	1,100	2,489	1,375	69.4	5.18	
11	E. B. Co.	4 V	16	635	40.0	7	8 1/4	1,150	5,090	1,580	86.0	8.0	
13	H.O.R.-M.A.N.	2DA	8	1,300	162.4	9.056	13.386	700	13,570	1,561	54.25	10.4	H.O.R. experimental
14	F-M Co.	4	4	37.5	9.4	4 1/4	6	1,200	340.6	1,200	72.7	9.08	
15	F-M Co.	4	8	120	15.0	5 1/2	7 1/2	1,200	1,425.6	1,500	55.5	11.88	
16	F-M Co.	2 O.P.	6	300	50	5	6	1,200	1,414.1	1,200	70.1	4.71	
17	Winton	4	3	37.5	12.5	5 1/4	7	1,200	454.7	1,400	54.4	12.11	
18	Winton	4	6	150	25.0	6	8	1,200	1,357.0	1,600	73.1	9.05	U. S. Navy standby generator
19	McIntosh & Seymour	4	6	120	20.0	5 1/2	9	1,200	1,283.0	1,800	61.8	10.60	
20	McIntosh & Seymour	4	6	150	25.0	5 3/4	9	1,200	1,402.4	1,800	70.6	9.35	
21	Cooper-Bessemer	4	8	150	18.75	5 3/4	7 3/4	1,200	1,610.1	1,550	61.5	10.73	
22	Cooper-Bessemer	4	8	105	13.13	5 1/2	7 3/4	1,200	1,473.1	1,550	47.1	14.06	
23	Navy DA	4	4	25	6.25	3 5/8	4 1/2	1,428	185.8	1,060	74.7	7.46	
24	Navy DB	4	6	60	10.0	4	5 1/2	1,700	414.8	1,560	67.4	6.91	U. S. Navy power boats
25	Navy DC	4	6	105	17.5	5 1/4	7	1,190	909.3	1,388	76.9	8.66	
26	Navy DD	4	6	105	17.5	4 1/4	5 1/2	1,885	468.3	1,728	94.2	4.46	

Abbreviations are used in the column "Type of engine" to designate the various builders as follows: Continental, Continental Motors Corporation; Cooper-Bessemer, Cooper-Bessemer Corporation; E. B. Co., Electric Boat Company; F.-M. Co., Fairbanks, Morse & Co.; H.O.R.-M.A.N., Hoover-Owens, Rentschler Company; M.A.N., Maschinen Fabrik Augsburg-Nürnberg; McIntosh & Seymour, Diesel engine division of the American Locomotive Company; Stearns, Frank B. Stearns; Sulzer, Sulzer Bros.; Sun Ship, Sun Shipbuilding & Dry Dock Co.; and Winton, Winton Engine Corporation.

exhaust valves in the head to obtain uniflow scavenging. This unit was based upon an 8 × 10-inch cylinder the design of which was by that time already far advanced. Test results of a single-cylinder experimental unit were sufficiently encouraging to permit the company to accept an order for two 8-cylinder in-line engines for the Chicago Century of Progress Exhibition which was to open in May, 1933. This engine, the data on which are given in line 8 of Table 2, was pioneering the use of a welded-steel housing. It was duly completed and successfully passed the prescribed acceptance tests as well as additional tests at the experiment station at Annapolis, Md. A similar engine of an improved design was installed at the Submarine Base, New London, Conn., for instruction purposes.

The unprecedented success of this type of engine in the railroad field is common knowledge today, culminating as it did in the erection by the General Motors Corporation of a brand new manufacturing plant at La Grange near Chicago for the construction of the streamlined locomotives including the Diesel engines driving them. May not the U. S. Navy claim some credit for this success?

Sun Shipbuilding & Dry Dock Co. contracted to construct a six-cylinder, two-cycle unit of the opposed-piston, two-crankshaft type, similar to the one used by Junkers in Germany for airplane propelling plants. A 6½ × 9¾-inch cylinder size which was successfully built in a heavier construction with side rods was chosen (Table 2, line 9). The engine passed modified acceptance tests without any stop whatever and showed the most remarkable fuel consumption of 0.354-pound per B.H.P.-Hr. It is now undergoing further service tests. Outstanding features of this engine are its aluminum frame, which caused much delay in its completion since five partially machined castings had to be rejected before one could be used for assembling, and its twin geared generators designed for a maximum speed of 2900 R.P.M.

The Continental Motors Corporation was commissioned to combine its experience in building radial airplane engines with the design of the so-called "Argyle" sleeve valve and develop a radial two-cycle Diesel engine with uniflow scavenging through ports controlled by this valve. A ten-cylinder 6½ × 7½-inch engine was designed to be mounted directly upon the bell-shaped housing of the generator. The unit passed acceptance tests at reduced rating and was installed for further testing at the New London Submarine Base. However, even at this reduced rating (Table 2, line 10) the company has every reason to be satisfied with its performance and is offering it commercially for locomotive purposes.

The Electric Boat Company preferred to build a four-cycle engine, equipped for supercharging, to satisfy the Bureau's requirements. A 16-cylinder, V-type, 7 × 8¼-inch engine to develop 635 B.H.P. at 1150 R.P.M. (Table 2, line 11) geared to an 1800-R.P.M. generator that was located directly below it, was contracted for. Outstanding features of this unit in addiiton to the speed and arrangement of the generator are:

1. Aluminum engine frame, which delayed this engine also and for the same reason, five partially machined castings had to be rejected before one could be used for assembly.
2. So-called Lanova cylinder head, or rather combustion space to permit the high brake mep.
3. Floating bushing over the crankpin to permit higher crankpin pressures and therefore shorter distance between centers of cylinders and lighter weight.
4. Geared supercharging blower to increase the mep rating.

Engine Corporation.

Many annoying minor troubles developed with this unit in addition to an exceedingly low priority which was habitually assigned to work on this unit by the management because of press of other more remunerative business. However, eventually all difficulties were overcome and the unit passed its official tests successfully and is now being held in readiness for further service tests.

Fairbanks, Morse & Co., although it originally declined to participate in the Bureau's program of development, shortly thereafter completed a new type of engine of the opposed-piston two-crankshaft type (line 16 of Table 2). This engine successfully completed the prescribed type-approval tests.

The Hooven-Owens, Rentschler Company, subsidiary of the General Machinery Corporation, also failed to take advantage of the encouragement given by the Bureau's development program, but later built on its own account a 1300-B.H.P. engine in accordance with the Bureau's requirements. This engine is of the M.A.N. double-acting two-cycle type, similar to the engines used in the German Navy. It successfully passed the type-approval tests developing 1300 B.H.P. at 700 R.P.M. in eight double-acting cylinders of 9.056-inch bore and 13.386-inch stroke (line 13 of Table 2). This engine is the only representative of the double-acting type and is also employing a welded-steel frame in one piece.

DIESEL ENGINES FOR DRIVING STANDBY GENERATORS.

A Diesel-driven generator is the logical solution of the problem to supply electrical energy in emergencies when the normal source of it fails to be available for some reason. A large number of high-speed units of several different sizes and makes have now been installed (see Table 2, lines 14 to 22). Both air and electrically started units have been developed.

These units are all elastically mounted to eliminate the effect of engine vibration on the ship's hull. Several different types of spring vibration damper have been used with success so that now elastic mounting is being specified even for the largest ship's service generator units.

DIESEL ENGINES FOR POWER BOATS.

The author has dealt in detail with the development of this type of engine in a previous paper and it is intended here only to summarize briefly the results. Forced by the necessity of eliminating the dangers of gasoline as soon as possible the Bureau purchased samples of all engines available in 1932. These were tested at the Annapolis Engineering Experiment Station on the dynamometer and in a specially constructed cold chamber to assure reliable starting under most adverse service conditions. Eventually the Bureau contracted with the Buda Company to supply a substantial number of engines of three different sizes and obtained a license for their reproduction at the Norfolk Navy Yard. (See Table 2, lines 23-25.)

Weight and space requirements of these engines exceed but little those of the gasoline engines that they replaced and the fourth size of engine (Table 2, line 26) that was developed recently is even lighter than a gasoline engine. All of them can be started with normal equipment at temperatures as low as 20 F. and with additional equipment even at 20 F. below zero. Legal restrictions forced the Bureau to develop the domestic manufacture of standard fuel-injection equipment and one of the first commercial fuel pumps produced by the Ex-Cell-O Aircraft & Tool Corporation successfully passed rigorous testing at Annapolis and was adopted. The original test pump is still in service together with several hundred others which have been

supplied since then. Lately two other commercial pump manufacturers established domestic production of fuel-injection equipment and these will be tested as soon as practicable.

As the result of the Bureau's insistence on weight reduction the Buda Company acquired the Lanova license, which permits a higher output with minor modifications of combustion space, and this resulted in the purchase of several previously mentioned DD engines, which will also be built at the Norfolk Navy Yard.

DIESEL ENGINES FOR AUXILIARY SHIPS.

To provide employment during the peak of the depression the Bureau prepared plans and specifications to re-engine several wartime steam-driven tankers but the program did not materialize. Need, however, for efficient auxiliary ships of the fleet has been growing as time goes on and it is now hoped that before this paper is delivered Congress will provide the authority and the means to proceed with the Navy's auxiliary program. In anticipation of the forthcoming need for providing a modern power plant for several types of ship of this program the Navy has secured funds and entered into contact with three qualified Diesel-engine builders to construct and test experimental three-cylinder engines of about 700 H.P. per cylinder. As the result of this action by the Navy the three world's outstanding designs of large-size double-acting two-cycle Diesel engines have been obtained, Burmeister & Wain by Nordberg Mfg. Co., M.A.N. by General Machinery Corporation, and Sulzer by the American Locomotive Company, and will thus become available for future needs of the Navy, the U. S. merchant marine, and other applications.

FUEL AND TORSIONAL VIBRATION RESEARCH.

With the foregoing commitments to the use of Diesel engines it is only natural that the Navy, coincident with the development of the programs mentioned, proceeded with the study of the type of fuels required for the proper operation of these new types of Diesel engine. Experimental equipment and qualified personnel have been assembled at the Engineering Experiment Station, Annapolis, and funds secured for a thorough and systematic study of fundamental properties of Diesel-engine fuels. Another paper before this conference is scheduled to deal with the results obtained by this activity of the Navy.

All engineers in any way connected with the development of Diesel engines are well aware of the effect of this malicious malady, torsional vibration, in retarding the growth of the industry. Without going into further details of the reasons why in the layman's mind the Diesel engine became the birthplace of torsional vibration suffice it to say that the Diesel engines suffered the most from it because the reciprocating mechanism of its predecessor, the steam engine, was not developed for the higher speeds and resultant multiplicity of torque impulses.

Torsional vibration thus becoming an obstacle in the way of progress with Diesel engines, the Navy proceeded to study it courageously and painstakingly. Many outstanding papers by naval personnel and by academic consultants engaged on naval problems have been presented in the last 12 years with the result that the high-speed Diesel engine of today need not suffer from the consequences of torsional vibration if its designers are willing to consult specialists or can spare time to study the subject.

INSPECTION OF NAVAL MATERIAL.

If any credit is to be claimed at all for the part that the U. S. Navy played in developing modern high-speed Diesel engines, the Inspection Service of the Navy must not be overlooked. The uncompromising attitude of this fine group of men, consisting of a permanent civilian force headed by commissioned officers of the Navy is frequently but little appreciated by the engine builder's management which places profits before future. Designers of new types of engine are usually unanimous in their appreciation of the Navy inspection which takes no cognizance of unwritten precedents or established shop practices if these are at variance with the requirements of the approved drawing, which represents the designer's concept modified, if need be, by the operating experience of the Bureau. Being thus independent of the shop and not subject to pressure by the management the inspection comes closer to the ideal than any industrially established system.

This review of basic facts was intended to present only a broad picture of the U. S. Navy sponsoring development of many types of Diesel engine, without going into details that would make the paper unwieldy and unread.

SUMMARY AND CONCLUSIONS.

Summarizing briefly, the following are the reasons why the U. S. Navy will keep on contributing to developing the best types of propelling machinery.

Necessity drives the U. S. Navy to outguess the world in the type of propelling machinery it uses. Being thus always dependent on new developments for its supremacy the personnel of the Navy develops the proper attitude for constructive progress. This attitude is reflected in Navy specifications wherein no limitations are prescribed if proof of performance is available. "Holding nothing holy" is the proper motto for research, but in the interest of economy, of time more than of money, do not neglect the facts established by others long before.

In conclusion the author desires to pay tribute to that large body of highly trained men afloat, both officers and enlisted men, who by their faith, enthusiasm, and intelligent nursing of the new types of Diesel engines make the largest contribution of the U. S. Navy. They establish public confidence in these new engines and thereby lay the foundation stones for their acceptance; it is used in the U. S. Navy.

WELDING AT WATERSIDE.

In the new high pressure Waterside Station in New York, the steam and water lines are arc welded using "chill rings." The joints are stress relieved in position by induced current and are inspected by X-ray methods. The following abstract is from an article of interest to all engineers by Mr. W. S. Morrison and Mr. W. J. Angus, published in Power, September, 1937.

High pressures combined with high temperatures require a piping joint free from service troubles. This fundamental was given careful consideration at Waterside Station of the Consolidated Edison Co. of N. Y., Inc., for its new 1200—1400-pound, 900-F. topping turbine and its two boilers. An elaborate study of various forms of joints showed that no form of flanged joint would stand up for any length of time under high-temperature service, substantiating the general opinion that a full welded joint is the practical answer.

Several designs of welded joints for high-pressure, high-temperature service were tested. Upset ends of pipe, fittings and valves specially machined to true dimensions, and fitted with a special backing ring recessed into the wall at the joint, appeared very good, but the expense was not thought justifiable. In the end, present general practice was followed, using the 45-degree bevel, butt-welded joint, Figure 1, for 2-inch pipe and larger, for wall thickness not exceeding ½ inch.

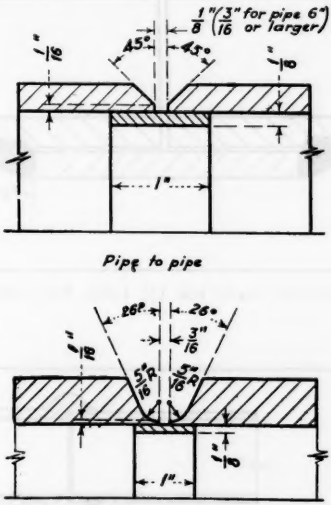

FIGURE 1.—JOINT (PIPE TO PIPE OR VALVE) FOR 2-INCH PIPE AND LARGER, IF PIPE WALLS ARE NOT OVER ½-INCH THICK.

FIGURE 2.—JOINT (PIPE TO PIPE OR VALVE) FOR PIPE WALLS OVER ½ INCH THICK.

All welding was done by d.c. electric-arc fusion method, which made it necessary to use a backing ring ("chill ring") to prevent formation of "icicles" or weld drippings and splatter inside the pipe. At first, we proposed to use a split ring of approximately the same material as the pipe, 2 inches wide and with a spacing rib (continuous or in sections) around the outer circumference at the center of ring width, but after experiments on trial welds, we selected a plain flat split ring 1 inch wide. For pipe smaller than 2 inches, where a backing ring is objectionable, fillet-socket joints were used, Figures 3 and 4.

MOST JOINTS WELDED.

As far as possible, all high- and low-pressure joints were welded on the first unit. Exceptions were made for boiler connections of the 1200-pound, 900-F. steam main, high-pressure boiler-feed piping, and the boiler-feed dis-

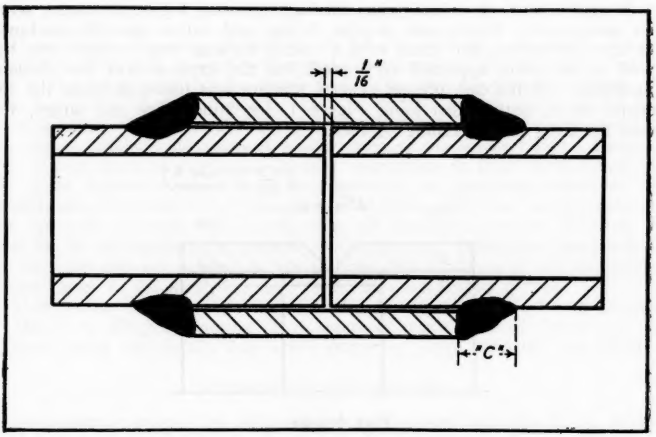

FIGURE 3.—SLEEVE JOINT FOR 1½ INCH PIPE AND SMALLER.

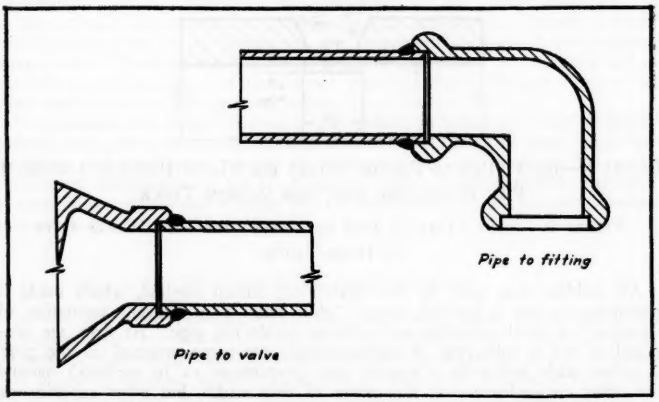

FIGURE 4.—FILLET-SOCKET JOINT CONNECTS 1½-INCH PIPE OR SMALLER TO VALVE OR FITTING.

charge at the pumps. For these connections, "Sarlun" ground-face flanged joints, without gaskets, were used. Other exceptions were made for connections to cast-iron valves and other equipment not provided with welding ends. Main steam piping from discharge ends of the non-return stop-and-check valves has welded joints throughout, including the connection to the turbine throttle valve.

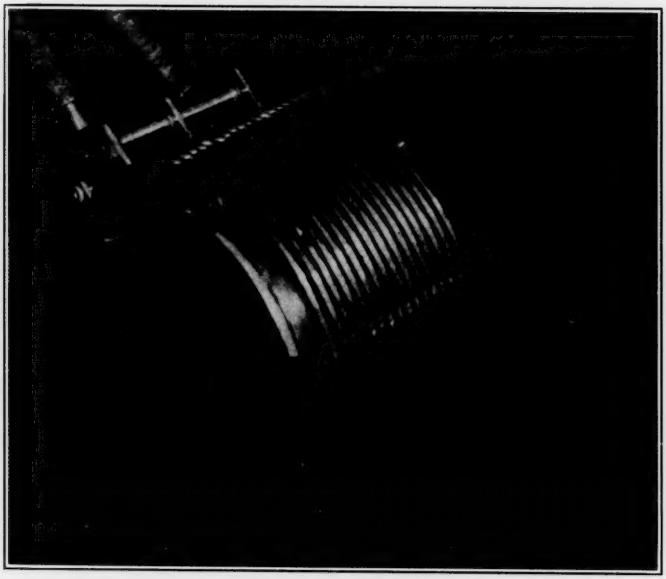

FIGURE 5.—STRESS-RELIEVING COLLAR, CLAMPED AROUND A WELD, ANNEALS WELD BY INDUCED CURRENT.

All stop valves on boiler-feed discharge lines (as on the main steam and the 200-pound bleed and station steam-tie lines) are welded directly into the line.

All shop fabrication and field welding of joints was performed by operators proven competent for the work involved by qualification under the rules of the Travelers Indemnity Co. Rules and qualifications are, in general, those of the American Tentative Standard Code for Pressure piping, ASAB-31-1935, with some additional features, such as requiring welders to make a branch connection.

In welding a butt joint in the field, the backing ring was inserted in position in one pipe end and tack welded. The adjoining pipe, valve, or fitting end was then brought into position and slipped over the free end of the backing ring, leaving the proper spacing between meeting ends. Four steel-bar lining-up strips were then tack welded to the meeting ends. These held the alignment until the weld was far enough along to permit their removal.

In laying the first bead, extreme care was taken to obtain thorough fusion at the bottom of the joint end and to fuse in all tack welds of the backing ring. To accomplish this without burning through the backing ring requires expert technique.

COATED ROD.

All welding was done with covered electrodes and a rod composition suitable to the piping material. A carbon-steel rod with the correct amount of molybdenum was used to weld the carbon-molybdenum alloy-steel piping. Small rods were used to lay in the first bead, since numerous thin beads gave a better weld than a few heavy beads. After each rod was used up and before subsequent beads were applied, a helper thoroughly cleaned each bead by light peening with a hammer and wire brushing.

Allowable mill tolerances (A.S.T.M.) in diameters, out-of-roundness and wall thickness of the pipe gave some difficulty in matching ends for welding. Spaces between pipe wall and backing ring resulted in slag inclusions and other imperfections in the weld. To obtain thorough fusion at the bottom of welds, it was sometimes necessary to machine special backing rings without a split, stepped in some cases, and fitted to the individual adjacent pieces being joined.

Main steam lines (carbon-molybdenum alloy-steel) and carbon-molybdenum valves in the boiler feed discharge lines required a special procedure to insure sound, ductile welds. Preheating to about 600 F., maintaining heat during welding, and completing the weld without interruption followed immediately by stress-relieving resulted in easier welding and a weld having better physical properties. A special resistance wire was wound around joining ends and covered with heavy plastic insulation. After the joint was completely welded, it was covered with additional insulation and current flow maintained for stress relieving. This procedure assured uniform preheating and maintaining of heat during welding. The insulation covering helps the parts to retain heat, protects the welder, and permits using the same equipment for stress relieving without change or interruption.

This procedure could not be followed in all cases because insulation could not be applied where the electric-induction stress-relieving equipment was used. In such cases, pre-heating was done with torches. Heat was maintained by the heat of welding and then the inductance collar was immediately clamped in place for the stress-relieving process. All welded joints on the main steam, boiler-feed discharge, and 200-pound bleed and steam lines, 2½ inches and larger, were stress relieved with the inductance collar, Figure 5, in practically all cases.

While it was found impossible to comply strictly with the Code of Pressure Piping in stress-relieving the weld within a band of the specified width, the procedure adopted was approved. With the inductance collar, maximum temperature in the pipe is at the center of the collar length, temperature falling off on each side. At a center temperature of 1200 F., temperature of a point 4 inches away is 1100 F. It is questionable whether a band as wide as the Code requires is necessary. After investigation, a practice was developed for keeping the weld and parent metal, to 1 inch each side of the weld, at a temperature of 1100 to 1200 F. for one hour per inch of pipe-wall thickness. This was done with both carbon and carbon-molybdenum steel, except that twice the time was allowed for " carbonmoly."

The stress-relieving inductance-type furnace was developed jointly by Detroit Edison Co. and its manufacturer, Detroit Electric Furnace Co. It consists of a split helical-coil collar, forming the primary of a low-voltage transformer to be clamped around the pipe, which then becomes the short-circuited secondary.

As soon as possible after completion of a weld, a collar of suitable size was clamped in place and the weld brought up to stress-relieving temperature.

The St. John X-Ray Service, Inc., under contract, made X-ray examination of all large welded joints on main steam, boiler-feed discharge, and 200-pound bleed and station steam-tie lines. Interpretation and responsibility was placed in the hands of the Service.

X-ray equipment used was unique—an extremely high-powered and highly mobile apparatus designed for use in close quarters under adverse conditions. The transformer and its water-cooling equipment are mounted on a trailer, then uncoupled and moved into the station to positions as close to the work as possible. 30 feet of flexible shock-proof, high-tension cable extending from trailer to X-ray tube made it easy to apply the tube in locations otherwise inaccessible.

The 220,000-volt X-ray tube, requiring about 8 milliamp., is especially designed for portability and pipeline welds. Ordinarily, tubes for stationary machines and boiler work are both oil cooled and oil insulated. Since the problem of maintaining oil insulation interferes with portability, this tube is water cooled and air insulated. The anode lead of the armored flexible cable encloses two concentric hoses conveying the cooling water to and from the tube.

The tube is mounted in a special shock- and X-ray-proof metal housing. It is designed for a point-source supply of rays and contains a filter to permit only the shortest penetrating rays to pass through the tube-housing window. This gives sharp definition and contrast in the negative. The machine is capable of penetrating 3 inches of steel in 1 minute with proper exposure. Exposures are made on a film contained within the lead-shielded holder, which is strapped in close contact with the pipe at that portion of the weld that is being examined and on the wall of the pipe opposite.

Exposures are taken in pairs, but not simultaneously or on the same film, and a sufficient number are taken to include the entire circumferential weld. Thus, for a 12-inch pipe, six pairs of exposures are taken. The tube is placed slightly to one side of the weld for one pair of exposures, and then slightly to the other side for the second pair, Figure 6. Rays are focused through the pipe wall adjacent to the tube, so that they pass through the entire weld in the opposite wall.

Satisfactory results were obtained and a number of poor welds detected. With these faults, weld metal was chipped out and the spot rewelded, stress relieved and re-examined by X-ray. Interpretation of exposures was based upon experience and knowledge of exposure conditions and weld design.

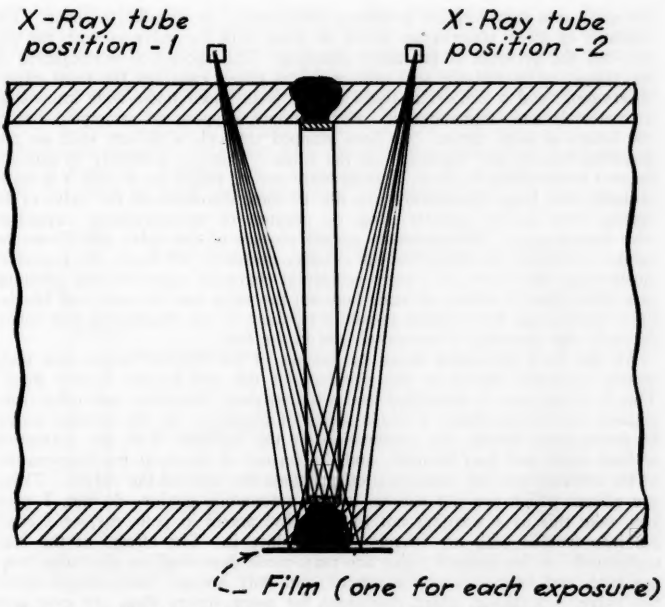

FIGURE 6.—EXPOSURES MADE AT ANGLE TO SHOW UP DEFECTS ALONG PLANE OF FUSION.

Angularity of exposures in pairs aids in the determination of fault position as well as its magnitude. Penetrameters or "tell-tales" were not used by the X-ray operator to determine depth of a fault. Instead, he used a set of comparative exposures on samples.

Because of the greater wall thickness (in this case 4⅝ inches), gamma ray was used for examination of one joint on the 20-inch main steam connection to the turbine throttle valve. Radium, centered and held in a special frame, was inserted through the disassembled throttle valve into the interior of the body and placed in line with the circumferential weld.

METALLURGY AND THE AERO-ENGINE.

Mr. D. R. Pye, in "The Engineer," September 10, 1937, discusses various metallurgical considerations in the selection of materials for aeronautical engines. Two subjects, namely, exhaust valve construction and friction surfaces are of such general interest to internal combustion engineers and to all machinery designers that they have been abstracted and are reproduced below.

The 100 per cent increase in the waste heat from the cylinder has profoundly affected the design and metallurgy of the exhaust valve in the last five years. It would surely be difficult to find anywhere so many thermal,

chemical, and metallurgical problems concentrated in so small a space. The difficulty of direct observation is not so great with the valve as with the piston, but the problem is far more complex. The successful development of the poppet valve and the seat onto which it closes calls for the most expert observation, and a critical analysis of the facts based upon wide knowledge of materials and engine design. For example, suppose an attempted run of 100 hours at high output has been stopped through a failure such as one describes loosely as "burning" of the valve. There is a variety of possible causes: overheating by itself, if sufficiently severe, might do it; but it is more probable that local overheating was due to some distortion of the valve or its seating ring in the cylinder head, on account of unsymmetrical expansion with temperature. This prevents perfect closure of the valve and allows an escape of burning gas under the full cylinder pressure. Or again, the imperfect closure may have been of a more accidental character caused by the pinching of a little piece of carbon or scale between the valve and its seat; and besides these mechanical and thermal possibilities, there is the chemical aspect introduced by the presence of tetraethyl lead in the fuel.

All the fuels demanded today by engines of the highest output owe their quality in some degree to the presence of the well-known "ethyl fluid." This is a mixture of tetraethyl lead with ethylene dibromide and other substances, and it introduces a whole series of problems for the exhaust valve. It decomposes during the combustion in the cylinder with the formation of lead oxide and lead bromide, and the former of these, at the temperature of the exhaust gas, has disastrous effects upon the steel of the valves. These are effects which are not serious in the automobile engine—do not, I pray you, take this as a warning not to use ethyl petrol in your car, or I shall get into trouble with my friends who produce it. The effects of the lead compounds on the exhaust valve are very much dependent on the valve temperature, and hence on engine output, and only become very serious when the valve is working under conditions far more severe than are ever met with in the automobile engine. Engine conditions, indeed, are extraordinarily critical. It has been found that a cylinder which will survive a hundred hours' running quite happily when developing a brake mean effective pressure of 180 pounds per square inch, will consistently fail through exhaust valve trouble when the power is increased by 10 per cent.

The most popular valve steel in use to-day is one containing 12 to 14 per cent each of nickel and chromium, but in spite of a high innate resistance to corrosion attack it is not able, by itself, to withstand oxidation in the presence of lead oxide at high temperatures. To get over this, advance has been along two lines: first, a complete re-design of the valve to maintain better cooling; and secondly, the covering of the seating ring, and now even the whole head of the valve, by stellite, a very hard alloy composed of cobalt, chromium, and tungsten, with about 2½ per cent of carbon, which has a better resistance to the effect of the lead.

The problem of cooling the valve is, like that in the piston, one of transferring the heat from the point where it is received—the head—to some point where it can be got rid of; ultimately, of course, to the surrounding air, but immediately to the metal of the cylinder body. With the mushroom valve there is little hope of getting rid of the heat directly from the head, and the problem is that of transporting it to the cooler stem, and thence across the working surface between the stem and the valve guide.

Twenty years ago, experiments were made at the Royal Aircraft Establishment on the removal of heat from the head to the stem by making the valve hollow and filling it with a fluid which was shaken up and down by the valve's motion. At first, mercury was tried, and later a eutectic mix-

ture of potassium and lithium nitrates. The suggestion was at that time premature, because the steel makers were able to produce better and better non-scaling steels which could work for long periods at a red heat, prior to the introduction of tetraethyl lead. In the last five years the introduction of the chemical problem has forced designers to means for getting better cooling, and to face the complication and expense of the hollow valve, containing metallic sodium as its heat carrier, which is now becoming universal in high-duty engines. In Figure 1 are shown cross sections of three slightly varying types of "liquid-cooled" exhaust valve. The sodium is solid when cold, but melts at about the temperature of boiling water.

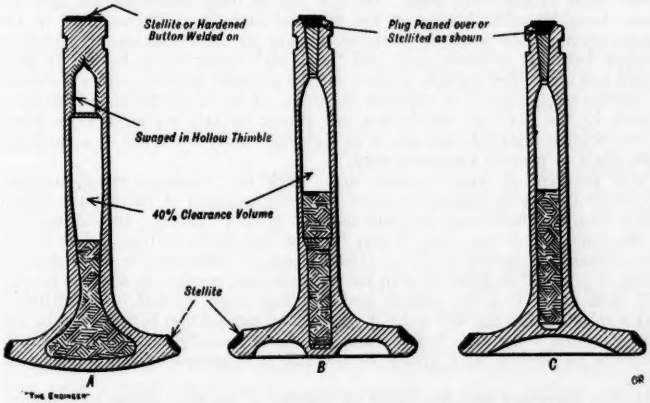

FIGURE 1.—DESIGNS OF LIQUID COOLED EXHAUST VALVES.

Even this complex design, however, combined with the stellite surface treatment has not sufficed to cure the troubles of the exhaust valve. There is evidence of a rapid increase in the rate of attack even on stellite, in the presence of lead oxide, between 600 degrees and 700 degrees C., and again above about 900 degrees C. The practice of covering the whole top surface of the valve, moreover, to prevent scaling, has led to trouble owing to a difference between the coefficients of expansion of the steel and the stellite which has been found to cause cracking of the stellite over the head surface, so that the covering has completely disintegrated after fifty or one hundred hours of high-duty running. A new material composed of about 80 per cent nickel and 20 per cent chromium has recently been tried for the same purpose and appears not to suffer in the same way when spread over the valve head. A possible further development is the fabrication of the entire solid parts of the valve from this non-ferrous alloy, thus avoiding the difficult welding processes involved in the composite valve. Finally, although the liquid sodium filling suffices to convey heat more rapidly from the valve head to the stem, it still remains to get this heat across the rubbing surface between the stem and the valve guide. Here also is an interesting problem, mechanical and metallurgical, which I shall deal with in the second half of my lecture.

Let me turn, now, to the problems where the emphasis is mechanical rather than thermal, although the temperatures at which things work are at

all times a vital factor. In this part of my lecture, I propose to leave aside for the most part the comparison of present and past designs and materials, and to consider some of the basic questions in physics and metallurgy which lie behind the engineers' problems of the future as I see them. Before I leave the practical engine, however, it is of interest to point out that the mechanical problem of maintaining two surfaces rubbing without seizure occurs in a variety of forms in the internal combustion engine. There is the crankshaft in its bearings, in the first place, and the piston in the cylinder; but equally important and interesting to the metallurgist is the problem of wear between the valve stems and their guides, and of the cam shaft and other parts of the valve gear. In each one of these directions the last five years have brought changes in the material and the technique used in the preparation of the working surfaces, always in the direction of providing greater hardness to resist wear and "pick up" under heavy loads. In both liquid and air-cooled engines we find, for the cylinder surfaces, either a change to special steels capable of extreme hardening or to chromium-plated surfaces. Similarly, the so-called lead-bronze has almost entirely replaced white metal as the bearing material, and this in its turn has involved the use of a hardened steel shaft to prevent excessive wear.

The problem of wear, seizure, and "pick up" between metal surfaces obviously cannot be considered apart from the question of lubrication, which under ideal conditions may preclude metallic contact entirely. Indeed, turning to the other end of the scale, it may be said that every surface, unless it has been chemically cleaned with the greatest care, is lubricated in some degree. Many of you will be familiar with the embarrassing manner in which a teacup will slide about in a dry saucer, and you may have learned by experiment that a splash of hot tea will make it stick. The explanation is that even in the best-regulated households it is normal for the supposedly clean, dry cup and saucer to be covered by a grease film, which is dispersed by the presence of water.

Having digressed into the realm of teacups, I am now going to continue the digression still further from everyday engineering and talk about metal surfaces in terms of molecules and molecular dimensions. Any surface, solid or liquid, is composed of molecules in motion, and a perfectly smooth surface would be one in which any irregularities were of molecular dimensions. We can picture the free surface of an undisturbed liquid as being smooth in that sense. It was the view of the late Lord Rayleigh that the difference between a liquid and a finely polished solid surface was not great, and that elevations on the solid surface would be of molecular dimensions. At the same time, it must be remembered that our finest test for the flatness, as distinct from smoothness, of a surface would only detect hills and valleys equal to a height of about a thousand molecules. While accepting Rayleigh's view, therefore, that the smoothness of a solid surface may approach that of a liquid, we may yet picture the most perfectly prepared plane surface as rather like a good golf green, smooth, but with ups and downs, and almost any bearing surface met with in engineering as being like a piece of smooth rolling downland. Figure 2 is intended to give some idea of what two supposedly flat surfaces, pressed together with an oil film between, would look like if sufficiently magnified. They touch only at the two points A and B.

When two surfaces slide past one another they may, in effect, be floating past one another on an almost continuous film of oil. This would be the state of things in a well-lubricated crankshaft bearing. On the other hand, the conditions may not be favorable to the maintenance of a continuous oil film. This is true between the piston and the cylinder, and more especially between the piston rings and the cylinder. In these circumstances the metal

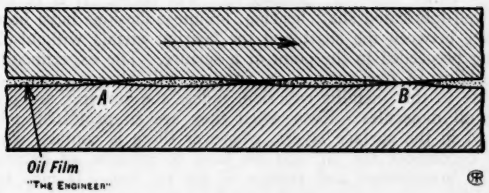

FIGURE 2—MAGNIFIED SECTION OF FLAT SURFACES IN CONTACT.

surfaces, as they slide past one another, come into contact, but only intermittently, and at points which are constantly changing. Remembering Figure 2, we can say that there is contact at the hilltops when these come opposite to one another, but that between-whiles the surfaces are separated by an oil film thick enough to contain many thousands of oil molecules. Thirdly, the surfaces may carry no visible lubricant at all, like the teacup, as may happen between a valve stem and its guide, or the load may be so concentrated, as on the working surface of a cam, that any thick film of oil is instantly squeezed away. I will mention, in passing, an interesting case of invisible lubrication between a steel valve and its bronze guide. It was on a well-known radial air-cooled engine. No special lubrication was provided for the valve stem, but it was quite able to run for some hundreds of hours without appreciable wear of the guide. Then some experiments were made, running the engine on a very economical fuel-air mixture, so that there was insufficient fuel to use up all the oxygen of the air, and it was found that the engine could not be run for more than twenty hours on account of excessive wear of the valve guides. The explanation appears to have been that under normal conditions a minute quantity of oil, or perhaps only oil vapor, was finding its way down by the valve stem and providing an invisible lubricating film. But when there was any free oxygen left over by the fuel, the oil vapor was burnt up and the wear increased by many hundreds per cent.

We must now touch upon what friction and wear really are and why they are reduced by a lubricating film. It used to be thought that the force of friction which opposes the sliding of one surface over another was always due to a sort of interlocking of minute asperities at the points of contact, however perfectly the surfaces might be polished. That view is now given up, and we believe that the resistance to motion is due to cohesive forces between the molecules themselves acting across the interface between the surfaces, and that these forces are of the same kind as the cohesive force between the molecules of a solid on which its strength depends. These cohesive forces between the molecules are very powerful at short range, but the range is extremely small. Remembering the "golf green" character of any smooth surface when considered in terms of molecular dimensions, it is clear that it is only over a small fraction of the apparent area of contact of two surfaces that the molecules will be brought within each other's range of attraction, and that owing to the short range of action of the cohesive forces even a single and quite invisible layer of oil molecules between two surfaces will suffice to prevent the molecules in each from getting to grips with one another. Whenever and wherever they do so, however, they will cohere with a force equal to that of the metal itself, and some molecules of one surface will be torn away, held firm by the molecules of the other; in

other words, either wear or "picking-up" will occur. It is of interest that even in a well-lubricated shaft bearing, minute adhesions have been observed to take place between the shaft and the bearing metals without these adhesions having proceeded to the point of a seizure. The experimental evidence is that a sliding of one surface on another will always cause some slight abrasion, due to this metallic cohesion, unless the surfaces are protected by a layer of oil molecules. Even when they are so protected, temperatures are produced in the surface layers, wherever there is contact, which are so high as to decompose the oil, and we have to imagine an extremely rapid succession of breakdowns and repairs of the oil film. It may be that the force of friction, once complete fluid lubrication has ceased, is due to the cohesion and parting again of innumerable pairs or groups of molecules, and that the observed value of the force is really an average result derived from a rapidly alternating state. If this is so, the likelihood of wear or of seizure must depend ultimately on the magnitude of the cohesive forces between the molecules on the two surfaces. Now the nature of these molecular forces which may be fatal to a bearing are essentially the same as the forces of chemical affinity, and just as the elements show a wide diversity in their eagerness to form chemical compounds with one another, so bearing surfaces must differ in their readiness to cohere when lubrication has failed. There are, of course, plenty of instances of two metals, A and B, which will work harmoniously together when A and C, under exactly the same conditions of load and temperature, will produce a seizure.

Our ability to preserve metal surfaces rubbing without seizure, in spite of heavier loading and rising temperatures, is vital to progress with the high-duty internal combustion engine, and I propose now to refer in more detail to the nature and behavior of surfaces and their influence on the engineer's problem.

So long as a complete unbroken oil film is maintained, this question of the proper "pairing" of metals does not arise, but even in a well-lubricated bearing there are the humps and hollows already described, and whenever two humps pass one another so that they are helping to carry the load on the bearing, then the oil is squeezed away until there is no more than a single layer or so of oil molecules between the surfaces; very high temperatures are produced, and intermittent failure of the oil film and metallic cohesions will occur. So long as this state persists, the two surfaces are said to be in the state of "boundary lubrication."

It is easy to understand why high temperatures of the metal surfaces make the danger of seizure more acute. In the first place the rate at which the oil is squeezed away from between the surface humps as they approach will depend on its viscosity. As temperatures rise the viscosity falls, the thick film of oil is more quickly dispersed, and the dangerous condition of boundary lubrication is earlier established. Besides this, however, there is the effect of a higher temperature on the danger of metallic cohesion when the oil film breaks down. At a higher temperature all chemical activity is enhanced, and there is therefore, *prima facie*, likely to be a direct effect of temperature in the direction of promoting metallic cohesion and seizure.

Accepting that between two heavily loaded surfaces in contact a condition of boundary lubrication, though intermittent, is constantly recurring and that, however good the oil, opportunities for metallic cohesion will occur, it is of great interest to consider what it may be possible to do to the metallic surfaces themselves to obtain relief and reduce the danger of seizure.

I have spoken already of the parallel between the forces of metallic cohesion and of chemical affinity and of the engineer's experience that some

metals show less readiness to seize together than others. I want now to speak of the light thrown by recent research work on the ultimate nature of surfaces, and how they may differ, and from that to suggest how the special treatment of surfaces may perhaps assist in the engineer's problem. The normal condition of a smooth metallic surface may be described as a crystalline aggregate with the atoms or molecules within each crystal arranged in a regular manner according to a "space-lattice" which is characteristic of the metal. On the surfaces of most liquids, on the other hand, the molecules, if we could see them, would appear completely without order of any kind, and there are many solids also which have this amorphous atomic structure.

Beilby—one of your Past-Presidents—after a careful microscopic study thirty-five years ago of the process of polishing, suggested that this did not consist simply of a rubbing away of asperities to smaller and smaller dimensions, but that the act of polishing produced a fundamental change in the character of the surface either of a metal or non-metallic crystal, during which the crystalline character is lost and the molecules become arranged "all higgledy-piggledy" in the way characteristic of a liquid surface. The correctness of Beilby's conclusion and the frequent occurrence of this amorphous layer, now known as the "Beilby layer," on a polished metal is generally accepted. It has received remarkable confirmation from the work of Bowden, and more recently from that of Finch and others with the electron camera. Bowden showed that when two metals were rubbed together, even quite lightly, temperatures could be recorded electrically at the surface which quickly rose to the melting point of one of the metals, and that by no increase of load and rubbing speed could the temperature be made to rise higher. He has since extended his observations more particularly to polishing, and finds evidence that the surface temperature, when a surface is rubbed with a polisher, always rises to the melting point of the polished surface. It must be understood, of course, that the very high temperatures are confined to an excessively thin layer on the surface, and do not spread perceptibly throughout the body of the metal.

An obvious deduction from these observations is that during the act of polishing, a liquid or semi-liquid layer of the metal is smeared over the underlying crystalline surface and, hardening again immediately, retains the amorphous character it had as a liquid.

Finch has confirmed this conclusion, but has further reached the extremely interesting and important one that this amorphous layer left by polishing may or may not remain amorphous according to the material polished and how it has been treated. I shall return presently to the practical importance of this conclusion. The thickness of the amorphous layer varies widely according to the material and the amount of polishing, from a layer only 20-30 atoms thick up to one of 100,000 or more. By examination of some new and some used aero-engine cylinders, Finch found that the "running-in" process between the piston and cylinder results in an extremely thick Beilby layer being formed; so thick that several rubbings with fine emery paper were necessary to remove the hard amorphous layer and re-expose the crystalline substratum.

It is, of course, common knowledge to the engineer that he must not allow an engine to give its full power until the "running-in" process is complete, or scoring and seizure between the piston and cylinder will infalliby occur. This conforms with the physicist's observations upon the Beilby layer, that metal in this amorphous condition is in general harder and tougher than the crystalline variety. Moreover, what is probably more important,

so long as the surface is crystalline one must expect its ultimate nature to be, as it were, a forest of minute crystalline peaks, rather than of the nature of a liquid at rest on which the surface irregularities are of molecular dimensions. It is fairly obvious that a metal surface of this latter type will be much more easily protected by an exceedingly thin oil film than a surface on which there are crystalline peaks each hundreds of thousands of atoms high, which will pierce the oil film and make metallic contact, with local high temperatures and an immediate danger, or even certainty, of some metallic cohesion.

Now let me return to Finch's observation that polished surfaces may or may not remain amorphous. The forces exerted by the atoms in the underlying crystal always try to rearrange the atoms in the amorphous Beilby layer, and in some materials they succeed, so that the polished layer is found to show some crystalline character under the electron beam. On other materials the amorphous layer shows a great reluctance to recrystallize. The forces of order exerted from the underlying crystal fail to bring about any rearrangement. I may mention in passing, as a matter of some interest, that the diamond is in a class by itself. Professor Finch was never able to detect the slightest trace of surface flow as a result of polishing. The polish on a diamond appears to be simply a rubbing down of the crystalline excrescences to smaller and smaller dimensions without their ever losing their typically crystalline character.

Diamonds seem a long way from the internal combustion engine, and yet it is quite possible these phenomena we have been discussing may prove of great importance if they lead to means for controlling the condition of the surface layer of metals used as bearing surfaces.

One of the substances which exhibited a stable amorphous layer after polishing was spinel, which is a magnesium aluminate with aluminium oxide in solid solution. Finch has suggested that if a suitably oxidized magnesium-aluminium alloy surface is used, it is possible by polishing this "to form spinel with a permanently amorphous and therefore smooth Beilby layer." The usual aluminium alloys used for pistons, on the other hand, become spontaneously covered with a thin layer of very hard aluminium oxide. This oxide layer is, or can be, made amorphous; but Finch's observations gave evidence that during the process of running-in the oxide layer, instead of settling down into a smooth amorphous bearing surface, became converted into a layer of minute sapphire crystals. It has long been known that the wear of the cylinder barrel is more severe with an aluminium than with a cast iron piston in spite of the greater softness of the former. It has hitherto been supposed this was due to the embedding of abrasive particles in the soft aluminium piston, but the explanation of Finch that aluminium forms its own peculiar kind of grinding surface with sapphire teeth, is not only more picturesque, but probably more true also.

Cast iron is an example of a material long known to be in the first rank as a bearing material, and here recent research has shown conclusively that it owes its quality to a unique faculty for preparing its own bearing surface. Cast iron contains minute particles of free carbon in the form of graphite. The methods of X-ray and electron diffraction have shown that the carbon atoms in graphite are built up together in the form of thin, flat plates or flakes, and that when a smooth surface of cast iron is polished, this has the effect of bringing out the occluded graphite flakes and of spreading them out over the iron surface so that their slip planes are parallel to it. In this way they act as a lubricating layer protecting the iron from abrasion. It is this unique power of forming a good bearing surface which keeps cast iron

supreme as a piston ring material in spite of its being entirely unsuitable from almost every other point of view. It has poor elastic properties, it is brittle, and it conducts heat badly; but the essential thing is that a piston ring must continue to rub without seizure under conditions in which satisfactory lubrication is impossible, and therefore we put up with the shortcomings of cast iron because of its peculiar virtue in this one respect.

A piston ring may seem an insignificant trifle in the whole design, but it is, perhaps, the most critical element in the piston, and this I have likened to the brain of the whole mechanism. I think I shall not be far wrong if I suggest that more money has been spent in bringing the modern piston ring to its present imperfect stage of development than on any other single element of the design, and the problem is still one in which there is great scope for the metallurgical physicist. The reason why piston ring development is so expensive is that nothing short of prolonged tests at nearly full load in the actual engine will search out its weakness, and the failure, if it comes, may be the starting point of widespread damage.

I have emphasized the importance of a knowledge of surface conditions and the ultimate structure of metal surfaces in terms of atomic arrangement, because I think I see there the necessary line of advance in the crucial problem of rubbing without seizure. Hitherto we have been able to rely safely upon an oil film to keep down the extent of the metallic contacts; but as the temperatures of the working surfaces increase the viscosity of the oil gives less and less protection, and the nature of the metal surfaces will play a more and more important *rôle* in the avoidance of seizure.

The line taken by the engineer today is to make use of one of the recognized methods, of which there are several, of producing a hardened surface; but do we know accurately, in terms of the ultimate atomic structure why one surface is hard and another soft? If the physicist would tell us that we might make a big step towards the goal of the perfect piston ring and the unseizable bearing surface.

FUELS OF TODAY AND TOMORROW.

This paper was presented by Mr. Arno C. Fieldner, Chief, Technologic Branch of the U. S. Bureau of Mines, as the presidential address of the Fortieth Annual Meeting of the American Society for Testing Materials, New York, June 28 to July 2, 1937. This comprehensive survey of fuels of the past, present and future was published in Combustion, August, 1937.

Thirty years ago coal furnished 89 per cent, oil 8 per cent and natural gas 3 per cent of the mineral fuel energy of the United States; water power contributed an equivalent of 3 per cent of the mineral fuels. After the War the displacing effect, first of petroleum then of natural gas, caused a steady decline in the proportion of energy from coal until 1933, the bottom year of the Depression, and at present coal furnishes about one-half of the total energy supply. This decline was augmented by the cumulative effect of research on increasing the efficiency of coal utilization.

RISE OF PETROLEUM.

The present age of intense competition of the different fuels has its underlying cause in the last century when Otto invented the gas engine in 1867 and Daimler built the first motor car using gasoline in 1887.

Thirty years ago an annual production of 166 million barrels of oil supplied the world with lamp and lubricating oils and was beginning to force its way to the front as a fuel for steam generation in competition with coal. By 1915 automotive needs for gasoline began to be felt. Prospecting spread to many States and the science of petroleum geology greatly helped in new discoveries. Following the War demand again overtook supply and, as a consequence, 34,000 wells were drilled in 1920 alone and a flood of oil from California and the mid-continent fields replenished the deficit.

In 1926 another shortage alarm appeared. The number of motor vehicles had increased to 22 million and the domestic gasoline supply was again falling behind the demand. New efforts were put forth and deeper drilling to depths exceeding 7000 feet, then to 10,000 feet and more recently to 12,000 feet, as well as the introduction of geophysical methods in finding oil strata, provided new supplies at an increasing rate. In addition, the enormous East Texas pool with 22,000 wells was discovered in 1930 by wildcat methods. Slightly over 1 billion barrels of oil were produced in the United States in 1929 and this figure was duplicated in 1936.

Parallel with the finding of new sources of crude oil, continuous progress was made in increasing the yield and quality of the gasoline produced. We have increased the production ratio of gasoline to crude oil from 13 per cent in 1912 to 44 per cent in 1935. The present average yield of gasoline by cracking is estimated at 60 per cent and each grade of fuel marketed throughout the broad expanse of the United States now has uniform properties.

Another landmark in the improvement of motor fuels was Midgley's and Boyd's discovery, inspired by Kettering, in 1922, that the addition of small quantities of tetraethyl lead and certain other substances to gasoline would suppress detonation and permit the use of higher compression ratios in the engine, thus increasing the efficiency of utilization. So great has been the bromine requirement of the lead anti-knock compounds that it became necessary for chemists to devise a process for extracting the minute quantities of bromine in sea water and a plant was built on the coast of North Carolina which now is extracting about 46,000 pounds of bromine per day from 145 million gallons of sea water.

DISPLACEMENT OF COAL BY FUEL OIL.

Liquid fuel was tried on ships before the World War and by the close of the War virtually all the navies were burning oil. The merchant marine followed and the oil-fueled tonnage of the world rose from 18 per cent in 1920 to 40 per cent in 1930 and to 50 per cent in 1936. Forty per cent of the oil-fueled tonnage in 1936 consisted of Diesel-powered ships. The railroads also have been and are today large consumers of fuel oil. While this has been limited largely to the Southwestern and Western States where coal is less available and fuel oil abundant, the later advent of Diesel-motored trains is encroaching on the use of coal in other regions.

The most recent trend is the ever-accelerating gain of oil heating for homes and buildings. The sale of heating and range oils increased from 25 million barrels in 1926 to almost 100 million barrels in 1936. One hundred million barrels of oil is equivalent to 24 million tons or one-half of the present production of anthracite. In all its uses the consumption of fuel oil alone is equivalent to about 100 million tons of coal—slightly more than one-fifth of the total production of anthracite and bituminous coal combined.

RISE OF NATURAL GAS.

Next to petroleum the most striking development in our fuel supply is the utilization of natural gas. Although employed for lighting Fredonia, N. Y., as early as 1821, the use of natural gas was sporadic until the discovery of reservoirs of gas in the search for oil. The Munroe field of Louisiana, discovered in 1916, indicated a reserve of 5 trillion cubic feet of gas which was enough to justify long distance pipe lines to centers of consumption. These were built in 1926 to New Orleans, Memphis, St. Louis, Birmingham and Atlanta. The largest reserve of all proved to be the Amarillo, or Panhandle field in Texas, estimates of which vary up to 41 trillion cubic feet of gas. Pipe lines from this field have been extended to Fort Worth, Denver, Kansas City, Chicago, Minneapolis and Detroit.

Fifty-five per cent of the natural gas produced and consumed in the United States comes to the surface of the ground in association with oil. From 0.5 to 10.0 gallons of gasoline and condensable hydrocarbons can be recovered from 1000 cubic feet of "wet" gas by compression and refrigeration or by absorption in oil.

COAL PRODUCTION AND UTILIZATION.

From 1898 to 1918 the production of bituminous coal increased at an average rate of approximately 20 million tons per annum, reaching a maximum of 570 million tons in 1918, which was the end of the rise. From that date the demand for coal wavered with the ups and downs of industry until in 1927 there began a decline which ended at a little over 300 million tons in 1932, and in 1936 production had returned to 434 million tons.

To hold the house-heating trade, coal producers are cooperating with appliance manufacturers in the development of automatic stokers which provide much of the convenience of gas- or oil-fired furnaces at a lower cost of fuel. The sales of domestic stokers have increased from 7000 in 1932 to 76,000 in 1936.

In the generation of power from coal the public utilities and the power plant equipment manufacturers have made great strides in improving fuel efficiency. From an average consumption of 7.05 pounds of coal per kilowatt-hour in 1899 and 3.2 pounds in 1919 to 1.44 pounds in 1936, is a truly remarkable record. Since the most efficient modern stations are producing 1 kilowatt-hour from 0.8 pound of coal, further decrease of the average may be expected as new equipment replaces the old. However, the best boiler furnaces are nearing their maximum possible efficiencies and further improvement of total fuel-power efficiency must be obtained in the conversion of steam to power. Metallurgical progress in the development of high-temperature alloy steels permits the use of higher steam pressures. The best record for a coal-steam-turbine plant is 10,900 Btu. per kilowatt-hour, or 31 per cent efficiency, which was obtained by the Port Washington Station (Milwaukee) operating at 1230 pounds pressure and 825 F. steam temperature. Mercury-vapor plants, still in the developmental stage, have given 32 per cent efficiency in regular operation.

CARBONIZATION AND GASIFICATION OF COAL.

Fifteen years ago the prospects of the coal-carbonization industry seemed very bright. The public was learning to burn coke and to appreciate its smokeless properties. The demand for gas was increasing and the supply of natural gas from the Appalachian fields was declining; the demand for

gasoline was mounting rapidly and discovery of new fields seemed inadequate. Further processing of coal for coke, gas and liquid by-products appeared inevitable; but unfortunately for the coal industry, the situation changed and a prolific period of discovery was followed by a veritable flood of oil and a hurricane of gas.

However, other factors also depressed the economics of coal carbonization. At present metallurgical coke is the only product from the processing of coal that does not have its price realization fixed by a competitive product. Light oil is controlled by gasoline; tar by fuel oil; ammonium sulphate by synthetic ammonia; and manufactured gas by natural gas. The commercial development of low-temperature carbonization, which provides an easily ignitable smokeless fuel, likewise has been retarded in the United States by the lack of demand for the light and heavy oils that are yielded in much greater volume than in the high-temperature process.

PRESENT-DAY FUEL SUPPLY AND DEMAND.

We have solid, liquid and gaseous fuels in abundance, and are using them at an increasing rate. The total energy supply from these mineral fuels increased from 14,000 trillion Btu. in 1907 to 24,600 trillion Btu. in 1929—a 75 per cent increase in 22 years. The Depression caused a drop to 16,132 trillion Btu. but recovery raised the figure to 21,800 trillion Btu. by 1936. The distribution of energy consumption in 1929, a year of high industrial activity, is given in Tables I and II. In round numbers, industry and public utility power consumed 45 per cent, transportation about a third and domestic and miscellaneous uses about one-fourth of our energy supply. Coal constituted 84 per cent of the fuel for domestic and miscellaneous purposes, 73 per cent for industrial use and public utility power and 57 per cent for transportation. Petroleum provided 43 per cent of transportation fuel, including practically 100 per cent of road vehicles and airplane fuel, 73 per cent of marine fuel and 11 per cent of railroad fuel. Natural gas comprised 16 per cent of industrial and public utility power consumption and 7 per cent of the domestic and miscellaneous needs.

During 1936 we produced and consumed 31 to 33 per cent of the world production of coal, 60 to 70 per cent of the petroleum and 95 per cent of the natural gas. We have a little over one-half (3.2 trillion tons of coal and lignite) of the world reserves of coal, and probably 60 per cent of the proved oil reserves. Two-thirds of the oil produced in all countries since its discovery in 1859 has come from American wells.

The deposits of oil shale, largely in the Rocky Mountain States, are estimated to contain a potential supply of 92 billion barrels of crude oil—an amount sufficient to maintain the present annual rate of oil production for nearly 100 years. The ultimate reserve of natural gas cannot be estimated.

The original reserves of coal and lignite are estimated at 3.2 trillion tons—enough to cover the entire state of Ohio to a depth of 76 feet. Not more than 2 per cent of the original supply of bituminous coal and about 25 per cent of the anthracite has been exhausted. At the 1929 rate of energy consumption, assuming that coal will carry the load after oil, gas and oil shale are exhausted, and allowing 30 per cent for loss, coal would last 2100 years. However, there certainly will be an increase in energy demand. An increase at the rate prevailing during the 'twenties would cut the foregoing period to some 500 years and a shortage of supply would be felt in the Appalachian field in the next 100 years. It must be remembered, how-

TABLE I

DISTRIBUTION OF ENERGY CONSUMPTION OF MINERAL FUELS IN 1929

	Coal, Million Net Tons			Petroleum, Million Barrels					Natural Gas	
	Anthracite	Bituminous		Gasoline		Kerosene,	Fuel and Gas Oil		Billion	
	Tons	Tons	Per Cent	Barrels	Per Cent	Barrels	Barrels	Per Cent	Cu Ft	Per Cent
Transportation	..	143	27.7	351	92.9	..	174	47	..	..
Railroads	..	135	26.2	1.1	0.3	..	76	20.5	..	..
Ships and boats	..	8	1.5	0.8	0.2	..	98	26.5	..	..
Road vehicles and airplanes	..	...	...	349.1	92.4	..	...	...	..	..
Public utility power	..	38	7.3	...	...	..	10	2.7	113	6
Public utility gas	..	6	1.3	...	...	..	22	6.0	..	..
Industrial	..	225	43.3	13.6	3.6	..	127	34.3	1444	75
Domestic and miscellaneous	69	106	20.4	13.2	3.5	36	37	10.0	360	19
Totals	69	518	100	378	100	36	370	100	1917	100

TABLE II

PERCENTAGE OF ENERGY SUPPLIED BY THE SEVERAL MINERAL FUELS FOR CERTAIN MAJOR USES IN 1929

Kind of Fuel	Domestic and Miscel.	Public-Utility Power, Gas and Industrial	Railroads	Ships	Road Vehicles and Airplanes	Total Trans.
Coal	84	73	89	27	...	57
Petroleum	9	11	11	73	100	43
Natural gas	7	16[a]	...	...	...	...
Totals	100	100	100	100	100	100

[a] Two-thirds of this 16 per cent used for gas and petroleum field operations, oil-refining fuel and production of carbon black.

ever, that the superior coals of the Eastern region comprise only one-seventh of the total reserves and one-half of the reserve consists of sub-bituminous coal and lignite, which are in the western part of the United States.

FUELS OF TOMORROW.

An appraisal of present trends and future probabilities indicates that:

1. Coal will continue to be the principal fuel used for the generation of public utility and major industrial power. Technologic improvements and new hydroelectric power will tend to reduce the consumption of coal. On the other hand, an increasing demand for energy and a decreasing supply of cheap residual oil will increase the amount of coal consumed for power purposes. No material change is expected in either direction in the near future, but in 10 or 15 years the trend will favor increased consumption of coal.

2. No substitute has appeared for metallurgical coke. The coke-oven industry will expand and consume more coal in accordance with metallurgical needs.

3. The convenience and uniformity of automatic heating of homes with gas or oil will continue to attract more users, even at higher costs than those prevailing today. Stoker-fired domestic furnaces, now in their beginning, eventually will give automatic service at a lower cost than for oil or gas, and high- and low-temperature cokes will supplement anthracite as solid smokeless fuel.

4. In 1929, 88 per cent of the railroad fuel was coal. Since then Diesel locomotives have been adopted by several railroads for light-weight, high-speed passenger trains and an increase in Diesels for such service is expected, but no general change in freight haulage from steam to Diesel power is likely to take place.

5. Marine transportation is energized by oil. Approximately three-fourths of the marine fuel used in 1936 was oil, and 40 per cent of this oil was used in Diesel engines. This trend will continue.

6. From the very beginning of the automobile industry recurring threats of shortage of gasoline were met by finding new pools and improving production technique, and in the refinery, by increasing yields and making a more efficient product. The end has not been reached. We are just beginning to use scientific methods in extracting oil from the sands and catalytic polymerization and hydrogenation eventually will furnish the means for complete conversion of volatile liquids and heavy petroleum to gasoline.

MISCELLANEA OF INTEREST TO NAVAL ENGINEERS.

WATER CIRCULATION IN BOILERS.—The Engineer, London, England, October 8, 1937.

The designer of water-tube boilers never admits any doubt as to the way the water circulates inside them. He will, if you ask him, be pleased to draw little arrows on the tubes pointing the way for the water, and he will add other curly ones here and there for the guidance of the flue gases.

Everything is then perfectly clear. Down goes the water from the drum by these tubes, across by those, and up by the others, round and round like the music and with equal persistence. The scheme would be convincing if only we could be sure that the water knew what the arrows meant. For the flue gases this does not matter, because they are bound to travel from the furnace to the stack by the only path open to them. They are not required to circulate; a single passage and their work is done. Circulation of the water, however, is essential to the proper working of an ordinary boiler. Not only is it conducive to rapid transmission of heat from the tubes to the water, but dead water in any tube is almost sure to lead to trouble. When stagnation does occur it is not necessarily because the water has no means of escape. It could leave freely by either end of the tube, but if the conditions are such that there is no inducement for it to prefer one direction of flow to the other, it just stays where it is. In a thick bank of tubes, for example, when those nearest the fire are designed to act as risers and those on the far side as downcomers, the water in the central rows may be doubtful as to which way to go, and consequently may be reluctant to move at all. The same state of affairs may be found in the intermediate tubes of a bank intended by the designer to consist of risers only, for it is common knowledge that on such banks there may be at times a reverse flow in the cooler tubes, owing to a circulation being set up within the bank itself. The evidence of defective circulation is severe pitting in the guilty tubes. Whatever the actual cause of the pitting may be, there is no doubt that in many types of boiler an experienced engineer can foretell almost exactly which rows of tubes are likely to suffer most from pitting, and it will always be found that these are the rows where inadequate circulation may be expected.

The causes of natural circulation are very simple and are, of course, well understood. The cooler portions of the water, being denser, tend to sink, displacing the more highly heated portions, the up-flow of which is considerably aided by the bubbles of the steam entangled with them. The action has been demonstrated by a thousand models and the mental picture of the process is always in the background of the designer's mind. But a glass model, working at or about atmospheric pressure, is a very different affair from an actual boiler. The one thing common to both, which is important, although not directly obvious, is that the rate of circulation is kept in check by its own consequences. Every increase in its speed tends to make the temperature of the contents of the boiler more uniform, while every approach to uniformity of temperature diminishes the forces on which circulation depends. There is thus a limit to the rate of natural circulation in any boiler, and this will be the lower the more nearly every part of the boiler is fulfilling its real function of producing steam rather than heating water. But there is another factor antagonistic to a brisk circulation which is becoming more evident than formerly. A modern power station boiler is often supplied with feed water preheated to a very high temperature. In the days when boilers worked at about 200 pounds, and were fed with water at 212 degrees or so, the temperature range in the boiler may have been nearly 180 degrees F. In a modern boiler working at 600 pounds pressure and supplied with feed water at 340 degrees, the temperature range will only be 150 degrees, with a corresponding reduction in the forces tending to promote natural circulation. More significant, however, is the effect of the increase in pressure. The bubbles constituting 1 pound of steam at 600 pounds pressure will displace less than 40 per cent of the water displaced by the bubbles of steam at 200 pounds pressure, and their effect in

causing circulation will be reduced in the same proportion. With boiler pressures of 1200 pounds and over, which are now common in the United States, the bubbles will displace only about half as much water as at 600 pounds, so that their effect is again reduced by half.

It would appear, therefore, as if the tendency of modern boiler practice was to reduce those factors which make for brisk circulation, while demanding at the same time average evaporation rates three or four times as great as engineers used to be contented with. Boilers are also becoming highly complex structures, with water tubes all round the furnace, and slag screens in addition, imposed on the original tube systems. When tubes exposed to furnace temperature are expected to evaporate 50 pounds or more of water per square foot per hour, it is obvious that a proper circulation must be maintained through them, and circumstances are combining to make it more and more difficult to ensure this by natural means. It is hardly surprising therefore to find schemes for producing a forced circulation through the tube system attracting the attention of many designers. The idea is by no means a new one. In the 'sixties of the last century quite a crop of forced circulation water-tube boilers was developed in the United States. The period was one of great activity in the development of water-tube boilers, and many of the designs were such that the water could hardly be expected to traverse the many sharp corners provided in its path unless it were positively pushed along. It may also be significant that the practice of pumping the water round the boiler coincided with a general increase in steam pressures, such as we are again witnessing to-day. But it soon fell into disuse, being found a needless complication when the boilers were so designed as to make natural circulation more easy. Whether history will repeat itself in this respect it is impossible to say. No one would want a circulating pump attached to his boiler if he thought he could do without one, for though the power it absorbed might not be worth considering, the need for its continuous operation at full boiler pressure and temperature would not add to the charms of the boiler house, even if it did not detract from the reliability of the plant. It is true that there are certain boilers now in use or in course of development in which there is no circulation at all in the true sense of the word, the water passing once only through the tube system as it does in an economizer. In such boilers the movement of the water is, of course, maintained by the feed pumps. They would be free from any difficulties due to stagnant water, but experience only will show whether a final solution is to be found in this direction. At present the overwhelming majority of boilers in service depend upon internal circulation, and the question is whether in the future this will be kept up by a natural or artificial means. If the tube system, including the water walls, can be so simplified that an adequate and continuous flow through all tubes can be assured without recourse to excessively high drums, there is little doubt that this is the better plan. Otherwise, the alternative of pumping the water round will have to be adopted.

INSPECTING WITHOUT DOING DAMAGE.—The General Electric Review, Schenectady, N. Y., October, 1937.

Vital parts of important machines, such as the bucket wheels of steam turbines, must be free not only from surface imperfections but also from voids and nonhomogeneous areas hidden in the interior of the metal. But any method used to detect such hidden imperfections must be nondestructive—must leave the parts uninjured and unchanged.

Such a nondestructive inspection method was developed in the Schenectady Works Laboratory of the General Electric Company many years ago and is still being used to advantage in several departments of that company. This method depends on the theory that any substantial change in the constitution or state of magnetizable material will be reflected in its magnetic characteristics.

A constant magnetic field is produced so that it penetrates through the entire thickness of a small area of the bucket wheel. The wheel is slowly revolved so as to change continuously the section within the field until the whole wheel is magnetically explored. Imperfections such as voids or nonhomogeneous areas in the interior betray their presence by producing disturbances in the magnetic field at the surface of the wheel.

Magnetic search coils are mounted close to this external surface. Any sudden change in the magnetic linkage, created by a discontinuity in the metal of the bucket wheel, produces a corresponding sudden electric voltage in the external measuring circuit. The imperfection is detected, and the imperfect bucket wheel is discarded.

ALMOST ELEMENTAL.—The Industrial Bulletin, Arthur D. Little, Inc., Cambridge, Mass., September, 1937.

The story of super-purity metals is a scientific serial with interesting installments appearing currently. The rated purity available in some of the well-known metals today is almost fantastically high; one of the latest is a lead given as 99.9999+ per cent fine.

Perhaps one of the most unusual properties of these super-pure metals is their high resistance to chemical attack. Minute traces of foreign elements are found to influence enormously their physical and metallurgical characteristics. It is also found that such properties as electrical conductivity and ductility of the highly refined metals are far different than formerly supposed. The narrow borderline between physics and the theory of metal structure may possibly be investigated by research upon them.

Super-pure metals have been used mainly on a micro-basis as sources of arc or spark line spectra in the spectrographic study of metals and metallic compounds; but as further research reveals uses in compounds, or coatings, and elsewhere, it may be that usage will increase materially. Zinc at 99.995 per cent fine is even a commercial possibility, and the 99.99+ material permits the making of zinc-base die castings definitely superior to those made with 99.9+ zinc.

The high degree of ductility of nearly pure zinc suggests the coating of wire and sheets of iron or steel to make a product superior in withstanding severe mechanical deformation, and to improve the reputation of galvanized or zinc-coated steel. This has led to the development of the Bethanizing electrolytic process of the Bethlehem Steel Company, wherein wire is plated directly from very pure solutions of zinc sulfate made from the leaching of zinc concentrate.

Aluminum, antimony, bismuth, cadmium, copper, iron, magnesium, nickel, silver, tin, gold, carbon and platinum have all been prepared in moderate quantities in the super-pure state by research laboratories of the metal and refinery companies. Information on many of them has been collected by Mr. Thomas A. Wright of Lucius Pitkin, Inc., for a special committee of the American Society for Testing Materials.

Distribution of super-pure metals is a problem at the present time. Many are available, but only experimentally, while cost is largely undetermined but is quoted extremely high. The National Bureau of Standards has been suggested as a distributing agency to take care of the growing demand for them for testing purposes and as standards.

Unconventional methods of metal purification have been developed which include novel forms of electro-deposition, sedimentation, resublimation, and crystallization. In super-refining a metal, it is not always considered necessary to start with one of the commercially refined grades; on the contrary, often a major impurity can be precipitated and carry with it all or a greater part of other allied impurities. The Westinghouse Electric and Manufacturing Company engineers, it is understood, refine their super-pure "Wemco" iron by a double open-hearth process. During these processes it is extremely important not to introduce impurities. Sodium, for instance, so common in ceramics, in perspiration and in the atmosphere along the coast, offers particular difficulty. Atmospheric dust must be avoided, as must impurities from the electrical contacts and solutions used in electro-chemical refining methods.

Tin is usually melted in a quartz or heat-resistant glass crucible and then cast in small bars using a porcelain combustion mold, when careful re-check must be made to determine whether constituents in the glass have contaminated the metal. Often a pure melt is poured directly into distilled water to make drops or a "moss" which for some purposes is even more convenient than small bars. For spectroscopic work, fine sawings or raspings of a metal are convenient; but when so prepared, it is imperative that the sawings be washed in acid to remove the adherent film of iron or iron oxide introduced from the use of the saw or rasp.

In the analysis of such metals, scientific tricks of chemical microscopy and fluorescence tests may often be used. Storage also introduces new problems in avoiding moisture and atmospheric impurities. Transparent wrappings employed for metals and carbons must be examined and carefully selected for freedom from contaminants. Handling by human hands must be carefully eliminated. Glass containers must be wear-resistant and tight, otherwise arsenic, zinc, boron, sodium, silica and calcium may be introduced. Some idea of the increased burden upon analysis methods may be gained by the impurities normal to the 99.9999 per cent pure lead, the greatest of which is iron at less than one part in a million.

DEAERATED FEEDWATER.—Power, New York, N. Y., September, 1937.

With modern methods of keeping boiler surfaces clean, corrosion and pitting is encouraged by presence of oxygen or air in feedwater, and it is important that all feedwater be deaerated. Spraying into a condenser where the vapor will give up most of its entrained air in a high vacuum is effective where flows are properly controlled.

Figure 1 shows the Rawlins system of handling condensate and makeup water. Where a comparatively small surge tank may be installed inside the plant building or otherwise near the feedwater heater, this system can be used successfully to deaerate all makeup water in the condenser, and handle it from there to the boiler, without allowing further contact with air. The surge tank must be closed, vented at the top to the heater.

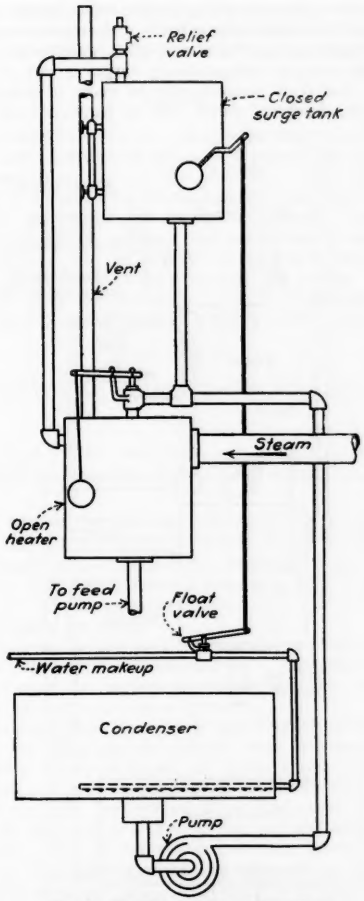

Fig. 1—Rawlins system of handling condensate and makeup feedwater

MAKEUP WATER.

All makeup water for the system is introduced through a perforated pipe inside the condenser. Makeup is regulated by a float valve with its controlling float in the surge tank. No other valves are required in the feedwater system, except the float valve regulating amount of water in the heater. Success of this system depends on having a makeup-water line and float valve designed to admit enough makeup water for the system, and

not too large for the condensate pump so that the condenser might be flooded. This system operates entirely automatically and requires very little attention, having no complicated valves, floats and other gadgets to get out of order.

Where the surge tank is comparatively large, is out of doors, or otherwise arranged so that it is impracticable to make it closed, as required by the Rawlins system, deaerated water may be handled as in Figure 2, which

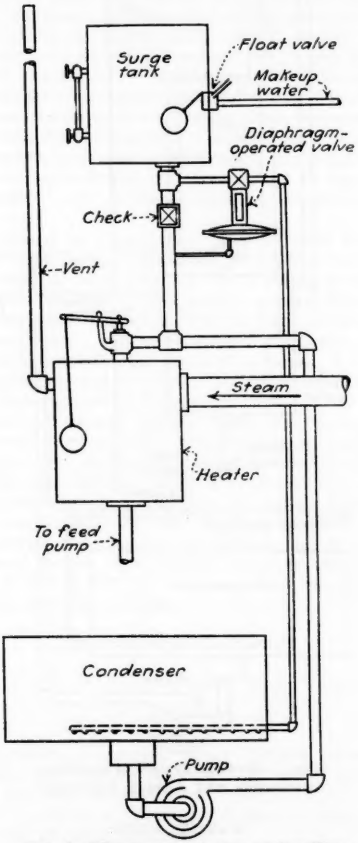

Fig. 2—Albuquerque system of handling condensate and makeup feedwater

we may call the Albuquerque system. With this system all makeup water goes into the surge tank, connected to the heater feed line through a check valve. Excess water in the heater feed line readily flows into the surge tank

through the check valve. Water cannot flow directly from surge tank to heater, but is admitted through a diaphragm-operated valve to a perforated pipe that sprays it into the condenser, where it is deaerated before going to the heater.

VALVES.

The diaphragm-operated valve is a single-diaphragm, spring-opened type, adjusted to stand open at pressures less than 1 pound gage on the diaphragm, and to close at pressures of 2 pounds or more. As in the Rawlins system, the arrangement for spraying water into the condenser must be properly designed so that it will admit the maximum makeup water required, and yet not be large enough to overload the condensate pump and flood the condenser. This system may be applied to feedwater entering an open or a closed heater. It is comparatively simple, not having any complicated mechanism and operating entirely with a single diaphragm valve, a check valve and a float-controlled, surge-tank, makeup-water valve.

COSTS FOR NEW DESTROYERS.—The Log, San Francisco, Calif., September, 1937.

Relative to the costs for the new destroyers to be built in the near future, the National Council of American Shipbuilders recently issued the following statement:

Increases in the costs in the new naval destroyers for which bids by private shipbuilders were recently opened are accounted for by the following reasons:

(1) These ships are of new design and somewhat heavier than the previous class on which bids were opened a year ago, with a resultant material increase in the cost of the ships themselves and also in the cost of the preparation of the plans for them.

(2) A sharp increase in the cost of material and equipment which the shipbuilders must buy from subcontractors. This increase amounts to about 45 per cent over the figures prevailing last year when the bids on the previous class of destroyers were prepared.

(3) Specifications for these destroyers require much more costly materials in many parts of the ships and in addition there is a requirement that the propelling machinery for them be set up in the shops, there given the full power test, then taken apart and installed on board the ships. This departure from previous practice adds very materially to final costs.

(4) Increases in the cost of labor in the private shipyards due to the operation of recent governmental regulations and legislation.

On the basis of estimates made by the Navy Department for building ships in navy yards it would appear that the cost of private shipbuilding is considerably higher than navy shipbuilding. This was indicated in the recent bids of the private shipbuilders for the two battleships as compared with the navy estimates.

On a fair basis of comparison, however, the private yards always have and doubtless will continue to build naval vessels at a lower cost than does the Government. Navy estimates are merely estimates. If a specific ship built in a Navy Yard costs one million, two million, or three million more than the navy estimate, the taxpayer has to pay the difference because in any event the ship has to be completed. On the other hand, a private builder submits a bid to the terms of which he is held by the Navy Department. If

the private builder is wrong in his cost calculations—and the private builders have been wrong in the case of a number of ships in recent years—he and not the taxpayers bears the loss.

It also is to be borne in mind that navy yards or the Navy Department in making up their estimates and in actually building the ships do not include as costs such highly important items as Social Security taxes, which in the case of the recent battleship bids ran as high as one million dollars, Federal, state and local taxes, maintenance of plant, and insurance; nor are the navy yards subject to the stiff penalties which are imposed upon private yards if they fail to complete the ships on time and if the ships themselves fail to meet various requirements as to speed, oil consumption, etc.

Likewise, it is to be borne in mind that the Government is fully protected from undue profit by private shipbuilders through the terms of the Vinson Act which sets a ceiling of 10 per cent on the profits which the private shipbuilders may make—a profit far below the figure upon which private industry generally in the United States is geared to operate.

On the other hand, there is no limit to the loss which the private shipbuilder may incur in the construction of a ship for the navy. Contrary to the impression which the Senate Munitions investigation attempted to create, heavy losses have been sustained by private builders in naval construction in both recent and long past years.

NOISE.—Engineering, London, England, September 17, 1937.

While it is still true to say, with Mark Twain, that "everyone talks about the weather but no one does anything," this does not in these days apply to noise, since it afflicts most of all the innocent, on the principle that we do not usually like a noise unless we make it ourselves. That the nuisance is of national importance is evident from the mass of information which goes to show that noise is one of the chief causes of fatigue in human beings, as is fully appreciated by those who return to city life after a period of years in quiet places. A probable reason is not far to seek, for sound is in general the consequence of mechanical motion which under certain conditions is transmitted to the auditory system, and it is reasonable to suppose that nervous energy is expended in the process. This is possibly conditioned by the fact that the sensation of sound is no longer experienced when the frequency of the disturbing source lies outside the approximate range of 20 cycles and 16,000 cycles per second. These are only average figures, the values of which commonly vary with the age of a person, who, with advancing years, may feel, rather than hear, the sound that is emitted by the 32-foot pipe of an organ. The subject may therefore be studied experimentally, on the assumption that the mental reaction can be expressed as a function of the mechanical phenomena. Although it is customary to make a distinction between musical tones and noise, implying that these arise in turn from periodic and non-periodic oscillations, it may be noticed that noise ensues from the simultaneous operation of a group of vibrations which separately are periodic in character. For instance, when as many piano keys as can be covered by the hand are struck at random, the noise which follows may be analyzed into components, consisting of perhaps eight or ten principal sounds, and each of these is a musical tone having a definite pitch-quality. Proceeding further, it would be found that each of the individual sounds involves a number of simple components, so that a hundred separate tones might be implicated in the single noise thus produced. But from this an ear could derive none of the sensations which we

associate with pitch and quality of tone. Hence it appears that the characteristic of a musical tone varies with the manner in which the simple components in a complex tone are distributed, and the resulting phenomenon is a noise when it lacks the coherence, continuity and pitch of musical tones. In other words, the nature of the boundary that separates these two classes of sound is psychological, and not physical. Notwithstanding, the subject of acoustics cannot be studied without recourse to the methods of physics.

SOME COMMENTS ON MICRURGY.—The General Electric Review, Schenectady, N. Y., October, 1937.

Ever since Leeuwenhoek in the middle of the 17th century made the first practical compound microscope and drew pictures of bacteria, that instrument has played a leading role in the development of the useful sciences. Curiously enough, however, the fact that many microscopic processes of biology, or more especially, of pathology, have their counterparts in inanimate matter has only recently led to the introduction of similar means of investigation into engineering practice. The laws of chemistry prevail in both fields, and since a chemical reaction obeys the same laws in the volume of a pinhead as it does in a tank we need only perform the necessary manipulation on a comparable scale and observe the result under appropriate magnification and illumination.

We term the operational part micrurgy—small work. It is something more than a mere reduction in scale of familiar laboratory practice mainly because the various phenomena of surface tension and adhesion, negligible there, here become dominant. While this acts as a handicap in some operations, it permits many others of remarkable effectiveness which have not before even been contemplated. These constitute perhaps the most characteristic feature of micro-chemistry.

Most reactions useful on the larger scale are useful also on the microscopic scale; but owing to the very large surface-volume ratios in the latter a whole new system, depending upon the rapid formation of microscopic crystals of distinctive form, become available.

The role of the new technique in our engineering practice promises to be chiefly in the field of "transformation products," hence the emphasis on the analogy to pathology. No sooner is the engineer's creation completed than its decay begins. The greater the stride into the hitherto untried, the more numerous are the unforeseen rifts in the fabric through which wear and exposure can advance to attack. And the tell-tale trail of the invader is nearly always microscopic. The composition of a speck of residual matter in the bottom of a tiny corrosion pit; of the film of "gum" on a defective bearing, whether of an electric clock or a turbine; of the faint discoloration on the bulb of a vacuum tube; of the smear of "dirt" on a contact; of a trace of process dust in the crevices of the windings of a burnt-out motor, determined by microchemical procedure, will usually lead to effective measures of correction with minimum of the wasteful process of trial and error.

THE NEW MOTOR TORPEDO BOATS.—The Shipbuilder and Marine Engine Builder, London, England, October, 1937.

The proof of the value of the new motor torpedo-boats as fighting units, it would appear, still belongs to the future, although for a decade these craft have been the subject of conjecture and fierce argument. One school boldly

asserts that they represent the answer to the big-ship problem, *i.e.*, they can hit and run, destroy the aircraft-carrier or capital ship, return to their base to refuel, and rearm ready to repeat the action. The other school contends that aircraft can and do accomplish all that is claimed for motor torpedo-boats, and with a greater degree of certainty. These diehards—among them many leaders of thought in the naval world—class these craft as frail experiments unsuited to arduous service at sea.

Despite these diametrically opposed opinions, it is certain that Germany, Italy and Russia (and possibly also Japan) have motor torpedo-boats of one form or another, but generally following similar classification.

The first motor torpedo-boat flotilla of H.M. Navy, comprising six "S.P." craft built to the designs of Mr. Hubert Scott-Paine and constructed by the British Power Boat Company, of Hythe, Southampton, accomplished the journey from Portsmouth to Malta under their own power and arrived one week ahead of schedule time. The actual times and performances are, of course, secret, but it is known that the craft were not pressed and could have reached their objective even earlier. No mechanical troubles of any kind were encountered, and they weathered a stiffish storm between Brest and Corunna, refuelling where necessary at sea—which of itself is interesting. Now they are apparently part of the Mediterranean Fleet, and no doubt we shall learn of their exploits in due course.

Mr. Scott-Paine (ex-airman and water-speed ace), who designed and built the units of the first flotilla of the new M.T.B.'s, must feel great satisfaction at this culmination of the task he set himself, and it is gratifying to Great Britain as a whole that once again she is first in the naval world as the possessor of a flotilla of such craft to serve in foreign waters.

Germany's motor torpedo-boats operate in German waters and have never ventured beyond the Baltic; Italy's are confined to the upper waters of the Dalmatian Coast; while Russia's and Japan's have not been to sea at all yet.

METALS WE EAT.—Mining and Metallurgy, New York, N. Y., October, 1937.

The necessity of an adequate supply of the bone-forming lightweight metals, calcium and magnesium, in the diet of animals and humans is not at all surprising to the nonmedical or even to the nonscientific mind. Probably also we are all accustomed to regard iron in the diet as essential to a normal and healthy condition of the blood. It may not, however, be equally well known, even to those quite familiar with the technologic uses of copper, zinc, manganese, arsenic, and aluminum, just what part these latter metallic elements play in animal nutrition. The following items are from a report of the Committee on Nutritional Problems of the American Public Health Association, published in the current Yearbook of the Association.

The earlier belief that only the iron present in foods in complicated organic form was absorbed and assimilated for hemoglobin production is no longer held. Recent work has shown that the value of different food sources of iron is correlated directly with the inorganic iron content. It is now generally agreed that copper is necessary for hemoglobin formation in human beings. Its role, however, is different from that of iron. Copper functions in the conversion of absorbed iron into hemoglobin. Figures suggested as representing a desired iron content of human diet vary from 0.6 milligram of iron per kilogram of body weight for children, up to 15 to 20 milligrams for adults.

The daily intake of copper varies from 0.8 milligram for a child's diet to 4.8 milligrams for a workingman's diet. Practically all foods contain available supplies of both iron and copper, but the fact that inorganic salts can serve as the sole source of these elements for hemoglobin formation has greatly simplified the treatment of anemia.

Evidence is accumulating to show that zinc is an essential element in animal nutrition. The average diet is not likely to be deficient in zinc since many foods contain a fair amount. Milk, for example, contains more zinc than it does either iron, copper, or manganese.

Manganese is widely distributed in both animal and vegetable tissues and may have an important part in the metabolism of both animals and plants but our present knowledge of its role is scanty. Likewise the question of a possible minimum requirement for arsenic in the animal body has not been answered.

The report concludes with the definite statement that normal amounts of aluminum are harmless when administered either in natural foods or when introduced by the use of aluminum utensils. This should help to lay at rest the query that continually recurs to or is planted in the popular mind as to the effects of the use of aluminum cooking utensils. The report says that there is no clear evidence that aluminum is an essential element in nutrition. Aluminum is, however, normally present in food supplies of both plant and animal origin.

The pathology which results from the addition of large amounts of soluble aluminum to the diet of experimental animals is a result of phosphorus depletion, essentially low-phosphorus rickets. Much the same type of interference with phosphorus assimilation results from the ingestion of large amounts of iron which similarly leads to rickets in growing animals.

BOOK REVIEWS.

MARINE ELECTRIC POWER. By Captain Q. B. Newman. Published 1937, by the Simmons-Boardman Publishing Company, 30 Church Street, New York, N. Y. 159 pps. Price $2.00.

In a very compact little book Captain Newman drives home in a non-mathematical and elementary manner the reason why electric generators generate power, and electric motors use it. As he assumes little basic knowledge on the part of his readers, and covers the field of direct and alternating current machinery most thoroughly, his task is a very difficult one. However, with a continuous use of homely analogies, and an elementary approach to each succeeding problem, he holds to his objective, and keeps his subject from getting out of hand. The book is recommended to those who have a good grade school and high school education, but who lack the time or opportunity to master the higher mathematics of electricity, and yet desire to know in a reasonable way why the wheels go 'round.

LANGE'S HANDBOOK OF CHEMISTRY. Second Edition. Published 1937, by Handbook Publishers, Inc., Sandusky, Ohio. 1800 pps. Price $6.00.

The second edition of a volume originally issued in 1934, compiled by Dr. Norbert A. Lange, Lecturer in Organic Chemistry at Cleveland College of Western Reserve University, assisted by Gordon M. Forker, of the Harshaw Chemical Company, Cleveland, Ohio, and containing an appendix of mathematical tables and formulae assembled by Dr. Richard S. Burington, Assistant Professor of Mathematics, Case School of Applied Science. The book has been extensively revised in its new form, and brought up to date to keep pace with the rapid strides in Chemical Engineering over the last few years. It is very handsomely bound, and has been well

printed in clear and readable type. It is tabulated and arranged especially for those requiring ready access to the chemical and physical data used in laboratory work and in manufacturing. It compares favorably in size, binding, and arrangement with the conventional handbooks of Marks, and of Kent, and will be found to be a desirable complement to them on the bookshelf of every engineer who has to do with chemical problems.

OIL BURNING. By Ir. H. A. Romp, M. E. Published in English at the Hague, by Martinus Nijhoff. 336 pps., 262 figures, and 7 tables. Price 12 Gld. (G = app. $.55).

The author has divided this rather extraordinary study of oil-burning under four main heads, as follows: I. The Historical Development. II. The Basic Principles. III. Modern Forms of Construction of Oil Burning Devices (complete through 1935), and IV. The Future Development. The book is a comprehensive and world-wide study of oil-burning apparatus carried up from about 1860 to the present time, in Russia, America, and Western Europe. Although foreign printed, the arrangement is quite conventional, the book is well printed with clear, readable type, and the text and illustrations are beautifully done. An unusual feature of the book is the use of trade names and manufacturers' names in the descriptions of apparatus and its development. It merits the earnest attention of engineers interested in the present and prospective uses of fuel oil.

ASSOCIATION NOTES.

ANNUAL MEETING.

The Annual Meeting of the Society was held on Tuesday, 5 October, 1937, at the Navy Department, Washington, D. C., with Rear Admiral Alfred W. Johnson. U.S.N., President, presiding.

The following were nominated for officers of the Society for the year commencing 1 January, 1938:

For President:

Rear Admiral H. G. Bowen, U. S. N.

For Secretary-Treasurer:

Commander R. W. Paine, U. S. N.
Commander T. A. Solberg, U. S. N.
Lieut. Commander R. W. Bruner, U. S. N.
Lieut. Commander H. L. Dodson, U. S. N.

For Member of Council:

Captain Albert Norris, U. S. N.
Captain H. M. Cooley, U. S. N.
Captain J. M. Irish, U. S. N.
Captain H. T. Smith, U. S. N.
Commander R. M. Griffin, U. S. N.
Commander C. A. Pownall, U. S. N.
Lieut. Commander J. C. Metzel, U. S. N.
Captain H. S. Howard (CC), U. S. N.
Commander H. A. Saunders (CC), U. S. N.
Captain H. F. Johnson, U. S. C. G.
Commander H. N. Perham, U. S. C. G.
Mr. B. W. Blair.
Mr. T. E. Cassey.
Mr. W. M. Corse.
Mr. F. J. Cunneen.
Mr. H. B. Gregory.
Mr. E. F. Sells.

Ballots have been distributed and polls will remain open until the close of business on 27 December, 1937.

It was voted to hold the annual banquet of the Society in 1938. Place, date and other arrangements to be in the hands of a Committee to be appointed later.

MEMBERSHIP.

There has been a most gratifying increase in membership during the year. Those joining since the publication of the August, 1937, JOURNAL are:

NAVAL.

Bird, L. G., Lieutenant, U. S. N. R., 126 South Cuyler Avenue, Oak Park, Ill.

Boyd, J. M., Lieutenant, U. S. Navy.

Broyderick, Leo J., Lieutenant, U. S. N. R., Naval Reserve Armory, Lynn, Mass.

Engstrom, Herbert L., Lieutenant, U. S. N. R., 2045 Anza Street, San Francisco, Calif.

Lemly, F. W., Lieutenant, U. S. N. R., 904 National Press Building, Washington, D. C.

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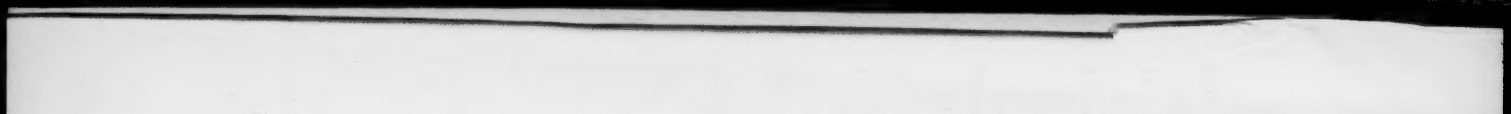
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